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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE

No. 1365

THE EFFECT OF CHANGING THE RATIO OF EXHAUST-VALVE FLOW CAPACITY TO INLET-VALVE FLOW CAPACITY ON VOLUMETRIC EFFICIENCY AND OUTPUT OF A SINGLE-CYLINDER ENGINE

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Massachusetts Institute of Technology

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THE EFFECT OF CHANGING THE RATIO OF EXHAUST-VALVE FLOW CAPACITY TO INLET-VALVE FLOW CAPACITY ON VOLUMETRIC EFFICIENCY AND OUTPUT OF A SINGLE-CYLINDER ENGINE

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SUMMARY

A series of tests has been made with a single-cylinder engine in order to determine the effect on volumetric efficiency and on engine performance of changing the ratio of exhaust-valve flow capacity to inlet-valve flow capacity when operating with exhaust pressure equal to inlet pressure. It was found that, within the range of speeds used, the engine gave best performance with values of the ratio of exhaust-valve flow capacity to inlet-valve flow capacity approximately equal to unity. This value corresponds to inlet and exhaust valves of approximately equal diameter and lift. It was also found that conventional valve timing gave better performance than any of the other timing arrangements tried, except at speeds higher than those used, where it appeared from the trend of the curves that delaying the inlet-valve closing would increase volumetric efficiency and mean effective pressures.

INTRODUCTION

One of the factors which limit the output of a four-stroke engine is the restriction to flow presented by the inlet and exhaust systems. The greater part of this restriction to flow usually is found in the valves and ports. Although other parts of the flow passages may also restrict the flow, these passages can usually be made larger to eliminate the restrictions. Increases in valve sizes, however, are limited because there is only a limited amount of space available for valves in the combustion chamber of a given engine and an increase in the diameter of the inlet valve can be made only at the expense of exhaust-valve size, or vice versa. Figure 1 illustrates this problem for the case of a typical aircraft-engine cylinder.

Analytical solution for the best compromise between inlet- and exhaust-valve size is extremely difficult. The best approach appears to be through experiment. An experimental procedure which was considered was to test various cylinders, each having a different ratio of inlet- to exhaust-valve diameter but being geometrically the same in all other respects. Such a procedure was rejected as both expensive and time consuming. Experiments in which the effects of changes in inlet- and exhaust-valve diameters are simulated by changes in inlet- and exhaust-valve lifts offered a quicker and less expensive means of investigating the problem, and this method was used in the present research.

A method of simulating changes in valve diameter by changing valve lift is suggested by the application of the orifice equation for an incompressible fluid to the flow through a poppet valve (see reference 1):

$$M = K D^2 C_{av} \sqrt{\Delta p}$$

where

M mass rate of flow

K a coefficient depending on units and on fluid density

D valve outside diameter

C_{av} the valve flow coefficient, averaged over the valve opening period

Δp pressure drop across the valve

It is evident from the foregoing expression that the average flow coefficient will vary as the valve lift is changed. In this investigation changes in the product D^2C_{av} were accomplished by changing valve lift, and it was assumed that the result, as far as flow was concerned, would be the same as if the same values of D^2C_{av} had been obtained by changing the valve diameter D. This assumption is justified by the experiments reported in reference 1, which showed that when an engine was operated with various combinations of inlet-valve diameter and lift, the volumetric efficiency could be expressed as a function of the following parameter:

$$\frac{\text{Piston speed} \times (\text{piston diameter})^2}{\text{Velocity of sound in inlet air} \times D^2C_{av} \text{ for the inlet valve}}$$

In the work of reference 1, the exhaust valve was large compared with the inlet valve, and therefore the chief restriction to flow was probably the inlet valve itself. Short inlet and exhaust pipes were used, as in the present tests, to prevent dynamic disturbances. Reference 1 showed that, as far as the inlet valve is concerned, it is immaterial to engine performance whether a given change in the product D^2C_{av} is produced by a change in valve diameter or by a change in average flow coefficient.

The average flow coefficient C_{av} (see reference 1) is obtained by making steady-flow tests through the valve and port at various lifts, at a pressure difference of 10 inches of water. These results are translated into a curve of flow coefficient against crank angle by means of the curve of valve lift against crank angle. The average flow coefficient is the average ordinate of the latter curve. The product D^2C_{av} is hereafter termed the "flow capacity of the valve."

Engine tests with varying valve diameter and lift, comparable with those described for the inlet valve, have not been carried out for the exhaust valve. It may be assumed, however, that the flow through the exhaust valve will also depend on D^2C_{av} if the flow coefficients are determined under the conditions of flow existing during the exhaust process.

The exhaust valve operates in both the sonic and the subsonic flow regions. Hu (reference 2) showed that for sonic flow in the exhaust valve the flow coefficients were essentially constant for cylinder pressures ranging from 28 to 80 pounds per square inch absolute when the back pressure was 14.7 pounds per square inch absolute. Later work ("Investigation of Exhaust Valve Design Using Steady Flow," by Harold J. Weiss and Yet Lin Yee, M.I.T. Thesis, B.S. degree, 1943) showed that the flow coefficients obtained by subsonic, steady-flow tests were in close agreement with the flow coefficients determined by Hu under sonic-flow conditions. It is concluded, therefore, that exhaust-valve flow coefficients are substantially independent of pressure and flow conditions, and that coefficients determined by subsonic, steady-flow tests at low pressure difference can be applied to the exhaust process. Thus, it seems reasonable to assume that the "flow capacity" may be used for the exhaust valve in the same manner as for the inlet valve in simulating the effect of changing valve diameter by means of the valve lift.

The investigation herein described was conducted to determine the best compromise between inlet- and exhaust-valve size for maximum air flow and power output. The objects of the investigation were (1) to determine the effect on air flow and power output of simultaneously ...ing the flow capacity of the inlet and exhaust valves of an ...e cylinder and (2) to determine the effect of changing inlet- and exhaust-valve timing on the air flow and power output of an engine operated with several combinations of inlet- and exhaust-valve flow capacities.

The tests were made at the Sloan Laboratories of the Massachusetts Institute of Technology under the sponsorship and with the financial assistance of the National Advisory Committee for Aeronautics.

DESCRIPTION OF APPARATUS

The engine used consisted of a typical aircraft cylinder of $5\frac{3}{16}$-inch bore mounted on a special single-cylinder crankcase (fig. 2). The stroke was 5.5 inches, and the compression ratio was 5.5. As shown in figure 1, both valves had an outside diameter of 2.218 inches. Special rocker arms were constructed which provided a means of changing the lifts of the inlet and exhaust valves (fig. 3). A special camshaft was used which has two convenient features: (1) the inlet and exhaust cams are separate and may be clamped in any position for the purpose of varying valve timing (see figs. 3 and 4); (2) the cams are easily removed and replaced with cams of different profile. A schematic diagram of the engine setup is shown in figure 5.

Air was supplied to the engine from a compressed-air line and was passed through an orifice meter and a surge tank into a steam-jacketed vaporizing tank, where the fuel was introduced. The resulting dry fuel-air mixture was then passed through a large, short inlet pipe to the inlet port of the cylinder. The exhaust gases were passed through a short pipe and a large surge tank into the laboratory exhaust system. The reason for using short inlet and exhaust pipes was to avoid effects due to inertia of the gases.

The cylinder cooling air was supplied by a large centrifugal fan, and special cylinder baffles were used. The cooling-air pressure was regulated by a valve at the fan inlet.

Lubricating oil was supplied under pressure to the engine by means of a separately driven gear pump. Oil was collected in the engine su and returned to the supply tank by a separately driven gear pump. brication was provided for the upper ball ends of the push ro of drip cups so located that the cooling-air blast carried onto the push-rod ends.

Measuring Instruments

Load was provided by an electric dynamometer. Torque was measured by means of a manometer connected to a hydraulically balanced piston attached to the dynamometer arm. Speed was determined by a mechanical tachometer in conjunction with a Strobotac operating from a 60-cycle alternating-current line and illuminating stripes painted on the flywheel.

Air flow was measured by a sharp-edge orifice installed in a 3-inch pipe line in accordance with the specifications of reference 3. Fuel flow was measured by a calibrated rotameter.

Spark advance was measured with a neon light, revolving with the crankshaft adjacent to a stationary protractor.

Standard laboratory equipment was used to measure temperatures and pressures.

Both heavy- and light-spring indicator cards were taken with the M.I.T. balanced pressure diaphragm indicator unit (reference 4).

Valve Timing

Valve timing was measured during actual running by means of electrical strain gages cemented to the push rods and connected through flexible leads to a 90-volt battery and ballast resistor. The variation in electrical resistance of the strain gage caused by changes in strain in the push rod caused a variation in voltage drop across the strain gage. This variation was amplified by the y-axis amplifier of a DuMont type 208 oscillograph. The resulting pattern which appeared on the oscillograph screen was a strain-time picture of the push-rod operating cycle which showed a clearly defined beginning and end. An electrical timing impulse was provided by a breaker operating at crankshaft speed and this impulse was superimposed on the push-rod strain pattern. By rotating the breaker, this impulse trace could be alined with the beginning or end of the push-rod strain pattern, and by using this impulse to flash the neon spark-timing light, the crank angle corresponding to the beginning or end of the valve motion could be measured.

By means of the timing apparatus just described, operating valve clearance was held at 0.045 inch at the cam during all tests. For purposes of computing the average flow coefficient, the valve-lift curve was based on the cam lift above the clearance line, as shown in figure 6. The opening and closing angles used to identify valve timing were taken at the intersection of the constant-velocity "ramps" and the cam-acceleration curves, in accordance with usual practice. (See fig. 6.)

VALVE FLOW CAPACITIES

In determining the combinations of inlet- and exhaust-valve lifts to be tested, it was decided to simulate, as far as possible, the design restrictions imposed by the engine cylinder. Figure 1(a) is a cross section of this cylinder, while figure 1(b) shows the head angles actually occupied by the outside diameters of the inlet and the exhaust valve, and the angles corresponding to the spaces between the valves themselves and between the valves and the top of the cylinder barrel.

With reference to figure 1(b), it was assumed that angles θ_1, θ_2, and θ_3 would be held constant for structural reasons. The two valves occupied a total angle $(\theta_4 + \theta_5)$ of 94°. In varying inlet- and exhaust-valve diameters, this 94° angle theoretically could be distributed between the two valves in any proportion. The resulting theoretical diameters over a reasonable range are shown in figure 7.

The average flow coefficients of the inlet and exhaust valves were determined from the results of the steady-flow tests (fig. 8) and the valve-lift and crank-angle relationships. The resulting curves of average flow coefficient are shown in figure 9.

Inlet- and exhaust-valve flow-capacity relationships are shown in figure 10. The space within the dotted rectangle of figure 10 represents the flow capacities that could be obtained with the apparatus. The curve A-B represents the variation in flow capacities that would have been obtained if the valve diameters had been changed in accordance with the values shown in figure 7, maintaining a maximum (valve lift/valve diameter) ratio of 0.225 for each valve. Because it was impracticable to run the engine with lifts higher than 1/2 inch, it was not possible to operate along curve A-B. The curve C-D was therefore chosen as a basis because it allowed a wide range of inlet-to-exhaust valve flow capacities without exceeding a maximum lift of 1/2 inch for either valve. Curve C-D represents the valve flow-capacity relationships in a geometrically similar cylinder head of smaller size than the cylinder actually used, but with available space for valves similarly utilized. Figure 11 shows the valve lifts required to give the valve flow capacities of curve C-D of figure 10.

METHOD OF TEST

Thirteen runs were made during the investigation. Data on valve lifts, flow capacities, and valve timing for these runs are given in table I. The runs have also been designated by number on figures 10 and 11 to permit a ready appraisal of the variations in flow capacities and valve lifts.

Runs 1 through 5 were made with valve flow capacities represented by curve C-D of figure 10. In these runs cams of 265 crank degrees nominal opening were used for both inlet and exhaust and the valve timing was unchanged.

Runs 6 and 12 were made at the same flow capacities as run 5, but with different valve timing for the inlet valve to determine whether an improvement in performance could be obtained. Similarly, runs 7 and 10 were made at the same flow capacities as run 1, but with different timings for the exhaust valve. Runs 11 and 13 were made at the same flow capacities as run 3, but an early exhaust opening was used for run 11 and a late inlet closing was used for run 13.

Runs 8 and 9 were made to show, along with run 3, the effect on engine performance when the exhaust- and inlet-valve flow capacities are changed, but the exhaust-valve flow capacity is kept equal to inlet-valve flow capacity. The same valve timings were used for runs 3, 8, and 9.

The following engine conditions were held constant during these tests:

Inlet pressure, in. Hg abs	33
Exhaust pressure, in. Hg abs	33
Inlet-air temperature, °F	120
Rear-spark-plug-gasket temperature, °F	450
Inlet-oil temperature, °F	150
Fuel-air ratio	0.075
Fuel	100-octane
Spark advance	best power
Compression ratio	5.5

For each setting of the valves and valve timing, readings were taken at various speeds from 500 to 2600 rpm.

RESULTS AND DISCUSSION

The results of the tests are presented in figures 12 to 23. The findings of principal interest are given in figure 12, which shows variation in volumetric efficiency with engine speed, and figure 13, which shows variation in indicated mean effective pressure, brake mean effective pressure, and pumping mean effective pressure with engine speed for runs 1 through 5 in which the inlet- and exhaust-valve flow capacities are changed along the curve C-D of figure 10. In the following discussion, the ratio of exhaust-valve flow capacity to inlet-valve flow capacity will be termed the "flow-capacity ratio."

In interpreting figures 12 and 13, it is necessary to make allowance for the fact that, in these tests, flow capacities of the valves, at any given flow-capacity ratio, were considerably less than they would have been if a series of cylinders of the same size and shape but with different valve diameters had been used, as suggested in figures 1 and 7. To assist in interpreting figures 12 and 13 as they would apply to a series of cylinders of the same size but with different relative valve diameters, a term "normal piston speed" has been plotted along the abscissa scale of figures 12 and 13.

By referring to figure 10, it will be seen that the curve A-B, for a set of cylinders with varying valve sizes, has valve flow capacities at a given flow-capacity ratio which are considerably higher than the flow capacities (curve C-D) of the cylinder used in these tests, when valve lift instead of valve size was varied.

Reference 1 indicates that, with given inlet and exhaust conditions, volumetric efficiency depends on the ratio of piston speed to inlet-valve flow capacity. Assuming this to be the case when both inlet- and exhaust-valve capacities are varied, it is evident that, for the same volumetric efficiency, the engine represented by the line A-B will run at a considerably higher piston speed than the engine represented by line C-D.

With the condition of equal inlet and exhaust valves as a reference point, the ratio of D^2C_{av} for the engine with variable valve diameters to D^2C_{av} for the engine with varying valve lifts is $\frac{1.54}{1.15} = 1.34$. Thus "normal" piston speed, or the piston speed of the engine with varying valve sizes would presumably be 1.34 times the piston speed of the engine with variable valve lifts to give the same volumetric efficiency with a given ratio of valve flow capacities. The normal piston speed plotted under figures 12 and 13 is, therefore, 1.34 times the piston speed at which the test engine was run.

Figure 12 shows that at normal piston speeds between 500 to 1800 feet per minute, variations in flow-capacity ratio have little effect on volumetric efficiency. Above 1800 feet per minute the curves separate, and the highest volumetric efficiency at high speeds was obtained with a flow-capacity ratio of 0.69. This ratio corresponds to a ratio of exhaust- to inlet-valve diameter of 0.83 if the two valves are of similar design. It should be emphasized that this result was obtained with exhaust pressure held equal to the inlet pressure. For a different ratio of exhaust to inlet pressure a different ratio of valve diameters for optimum volumetric efficiency might be obtained. It is also noticeable that the flow-capacity ratio of 1.00 gives nearly the same volumetric efficiency at all speeds as the flow-capacity ratio of 0.69.

Figure 13 shows brake mean effective pressure as measured by dynamometer torque and indicated mean effective pressure and pumping mean effective pressure as measured from the indicator cards. This figure shows that, in the range of normal piston speed between 1000 and 2000 feet per minute (this range includes normal cruising piston speeds at the present time), the various valve arrangements make little difference in the brake mean effective pressure. Above this speed, the highest brake mean effective pressure is obtained when the valves have a flow-capacity ratio of 1.00.

Figure 13 also shows that indicated mean effective pressure at high speeds is highest with a flow-capacity ratio of 0.69. This agrees with figure 12, which shows that a flow-capacity ratio of 0.69 gives the highest volumetric efficiency at high speeds. The difference in flow-capacity ratios giving maximum indicated mean effective pressure and maximum brake mean effective pressure seems attributable to higher pumping losses with the lower flow-capacity ratio, which is confirmed by the pumping mean effective pressure curves in figure 13. The probable error in measurements of indicated mean effective pressure and pumping mean effective pressure from the indicator cards is rather large, however, as indicated by the evident discrepancies shown at 1500 rpm in figure 13. It is therefore recommended that the curves of indicated mean effective pressure and pumping mean effective pressure in this report be interpreted with caution.

It is also quite possible that the small differences between the curves for flow-capacity ratio 1.00 and flow-capacity ratio 0.69 in both figures 12 and 13 are within experimental error, in which case it may be concluded that changes in flow-capacity ratio between 0.69 and 1.00 have little effect on performance.

Figures 14 to 21 show results of tests made to determine the effects of several changes in valve timing. The "normal" timing, which was used in all runs plotted in figures 12 and 13, was as follows:

Inlet opens	19° B.T.C.
Inlet closes	66° A.B.C.
Exhaust opens	72° B.B.C.
Exhaust closes	14° A.T.C.

Figures 14 and 15 give the results of runs 1, 7, and 10, in which, with a flow-capacity ratio of 0.51, three different exhaust-valve timings were used, the timing in run 1 being normal. Run 10, in which the exhaust valve was opened 34° earlier than for run 1, with other valve events unchanged did not result in any significant improvement in volumetric efficiency, indicated mean effective pressure, and brake mean effective pressure. Run 7, however, shows that volumetric efficiency, indicated mean effective pressure, and brake mean effective pressure were greatly lowered when both the opening and the closing of the exhaust valve were made to occur 25° earlier.

Figures 16 and 17 show that, when the engine is operated with a flow-capacity ratio of 1.00, opening the exhaust valve 34° earlier than normal, with other valve events unchanged, does not result in any notable improvement in volumetric efficiency or indicated mean effective pressure. The brake mean effective pressure values are lowered with the early exhaust-valve opening.

The effects of changing inlet-valve timing with a flow-capacity ratio of 1.79 are shown in figures 18 and 19. As compared with the normal timing of run 5, the inlet valve was opened and closed 27° later in run 6; whereas for run 12 the only change in valve timing was to close the inlet valve 35° later. Both of these changes in inlet-valve timing resulted in lower volumetric efficiency, indicated mean effective pressure, and brake mean effective pressure, in the lower speed range. The differences in volumetric efficiency, indicated mean effective pressure, and brake mean effective pressure for the three valve timings became less as the speed was increased. Above 2100 rpm (corresponding to a normal piston speed of 2500 ft/min) the timing used in run 12 results in a slightly higher volumetric efficiency. The trend of the curves in figure 19 suggests that above 2400 rpm (corresponding to a normal piston speed of 2850 ft/min) the timing used in run 12 would give an increase in indicated mean effective pressure and brake mean effective pressure.

Figures 20 and 21 show that, when the engine is operating with a flow-capacity ratio of 1.00, closing the inlet valve at 101° A.B.C. instead of at 66° A.B.C. results in lower volumetric efficiency, indicated mean effective pressure, and brake mean effective pressure, over the speed range investigated. As the engine is operated at higher speeds, however, the differences become less. The trend of the curves indicates that, at speeds above the highest used in this investigation, later inlet-valve closing may give higher volumetric efficiency, indicated mean effective pressure, and brake mean effective pressure than the normal timing.

Figures 22 and 23 show the effects of changing the flow capacities of the exhaust and inlet valves while maintaining a flow-capacity ratio of 1.00. The decrease in volumetric efficiency, indicated mean effective pressure, and brake mean effective pressure at the higher engine speeds as the flow capacities are reduced reflects the curtailment in maximum engine performance when the engine designer does not utilize the maximum flow capacities possible for the inlet and exhaust valves. In practice, low valve flow capacities could result from using small valve diameters, low valve lifts, or poorly designed valves and ports. (See reference 5.)

INDICATOR DIAGRAMS

Light-spring indicator diagrams taken during these tests are reproduced in figures 24 to 38. Superposition of the individual diagrams was based on the common inlet and exhaust pressure line ($p_i = p_e = 33$ in. Hg abs.). There is evidence that some of the diagrams, in particular those of figure 36, are displaced in the pressure direction, indicating an error in the location of the "atmospheric" line on the original indicator cards. Caution, therefore, is advised in comparing the various diagrams quantitatively.

In general, the diagrams show expected trends, that is, that pressure losses through the valves increase with increasing speed and with decreasing flow capacity. The figures furnish interesting data on the detailed effects of the various valve-lift and valve-timing combinations.

CONCLUSIONS

Tests made with a single-cylinder engine to determine the effect on volumetric efficiency and on engine performance of changing the ratio of exhaust-valve flow capacity to inlet-valve flow capacity show that, within the range of the tests and when inlet pressure is maintained equal to exhaust pressure:

1. A flow-capacity ratio of approximately 1.00 gives highest brake mean effective pressure over most of the speed range. This flow-capacity ratio corresponds to equal diameters for exhaust and inlet valves of similar design.

2. Reducing the flow-capacity ratio from 1.00 to 0.69 effects only a small reduction in brake mean effective pressure - possibly within the experimental errors involved.

3. Highest volumetric efficiency over the speed range was obtained with a flow-capacity ratio of 0.69. Increasing this ratio to 1.00 gave only a slight reduction in volumetric efficiency - possibly within experimental error.

4. Tests with several arrangements of inlet- and exhaust-valve timing showed no significant improvements in volumetric efficiency, indicated mean effective pressure, and brake mean effective pressure over the values obtained with normal timing (inlet opens 19° B.T.C. and closes 66° A.B.C., and exhaust opens 72° B.B.C. and closes 14° A.T.C.) within the range of speeds actually used in the tests. The trend of

several of the curves showed, however, that improvement in output at higher speeds should be expected with a delayed inlet-valve closing.

Employment of ratios of exhaust to inlet pressure not equal to 1 might result in a modification of these conclusions.

Massachusetts Institute of Technology,
Cambridge, Mass., April 25, 1946.

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TABLE I.- SCHEDULE OF VALVE LIFT, VALVE TIMING, AND FLOW COEFFICIENTS FOR ALL TESTS

Run	Valve lift (in.)		L_{max}/D		C_{av}		D^2C_{av}		Flow capacity ratio	Valve timing			
	Inlet	Exhaust	Inlet	Exhaust	Inlet	Exhaust	Inlet	Exhaust		Inlet opens (°B.T.C.)	Inlet closes (°A.B.C.)	Exhaust opens (°B.B.C.)	Exhaust closes (°A.T.C.)
1	0.500	0.244	0.225	0.110	0.319	0.163	1.569	0.802	0.51	19	66	72	14
2	.405	.290	.183	.131	.281	.193	1.380	.948	.69	19	66	72	14
3	.333	.358	.150	.162	.234	.235	1.151	1.155	1.00	19	66	72	14
4	.284	.428	.128	.193	.197	.273	.969	1.343	1.39	19	66	72	14
5	.248	.500	.112	.225	.170	.305	.836	1.500	1.79	19	66	72	14
6	.248	.500	.112	.225	.170	.305	.836	1.500	1.79	[a]8	94	72	14
7	.500	.244	.225	.110	.319	.163	1.569	.802	.51	19	66	97	[b]12
8	.459	.500	.207	.225	.305	.305	1.500	1.500	1.00	19	66	72	14
9	.244	.250	.110	.113	.167	.167	.821	.821	1.00	19	66	72	14
10	.500	.244	.225	.110	.319	.163	1.569	.802	.51	19	66	106	14
11	.333	.358	.150	.162	.234	.235	1.151	1.155	1.00	19	66	106	14
12	.248	.500	.112	.225	.170	.305	.836	1.500	1.79	19	101	72	14
13	.333	.358	.150	.162	.234	.235	1.151	1.155	1.00	19	101	72	14

[a]Intake opened 8° A.T.C.

[b]Exhaust closed 12° B.T.C.

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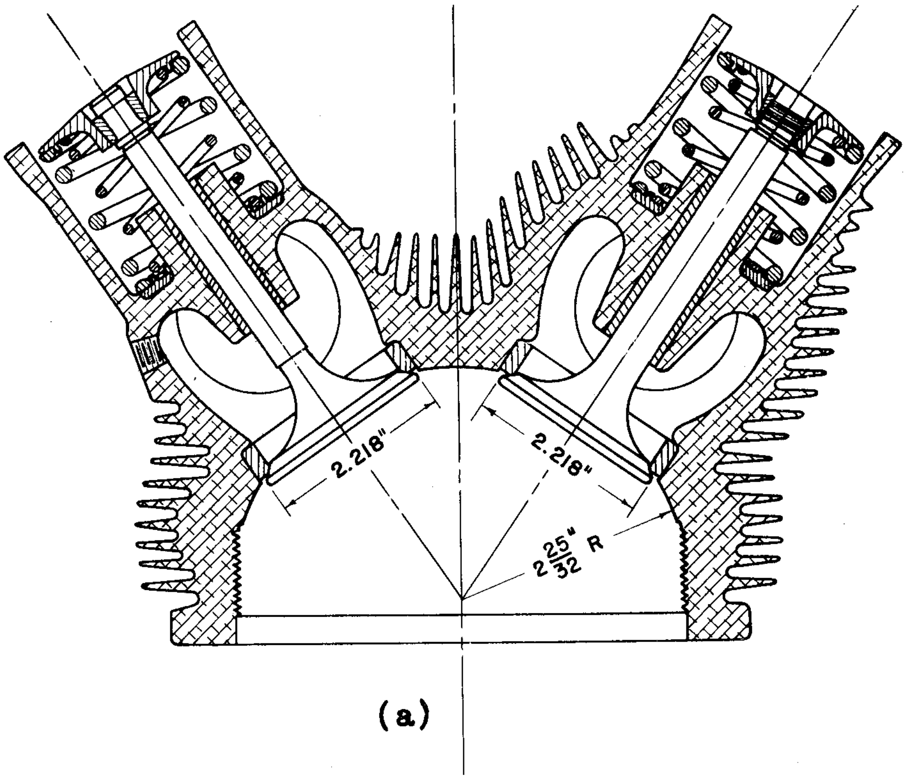
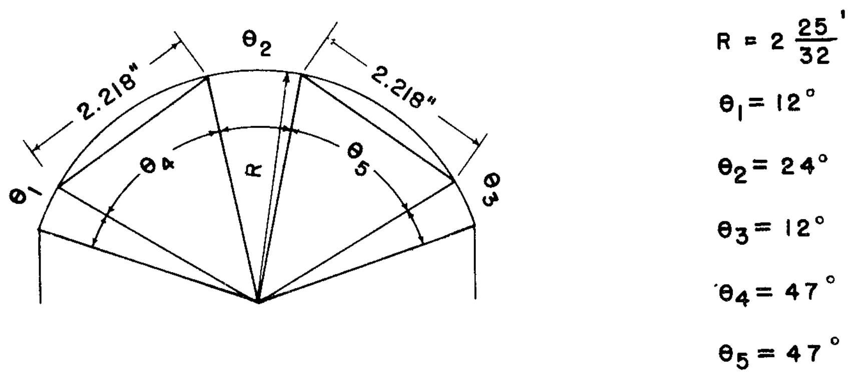
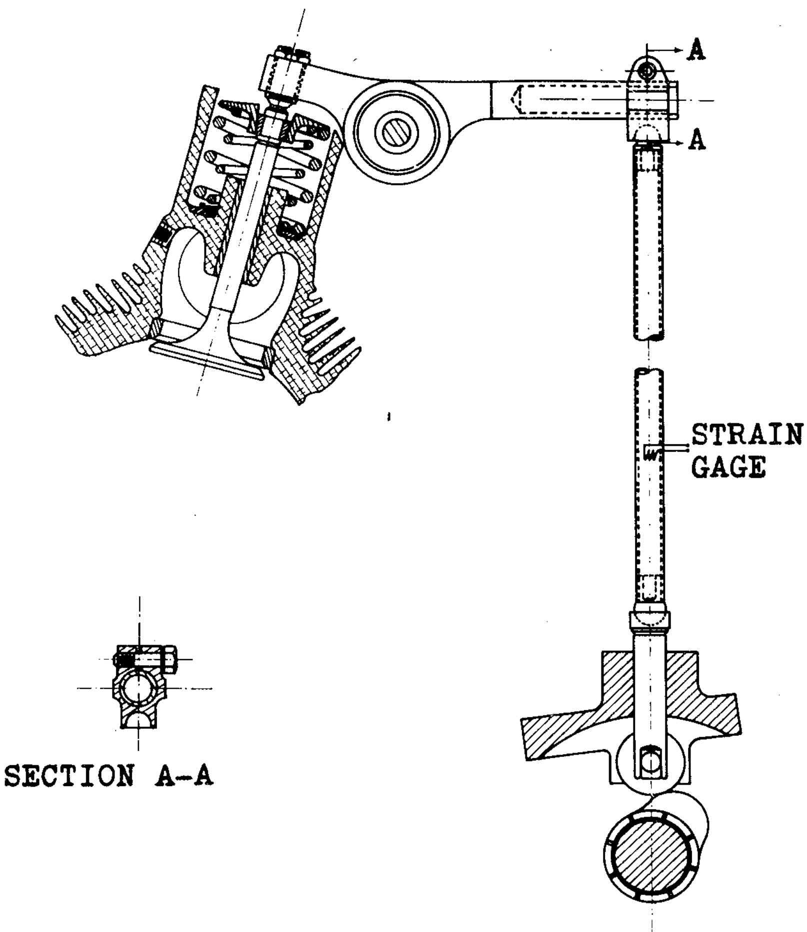


(a)



(b)

Figure 1.- Geometry of valve space requirements in an engine cylinder.



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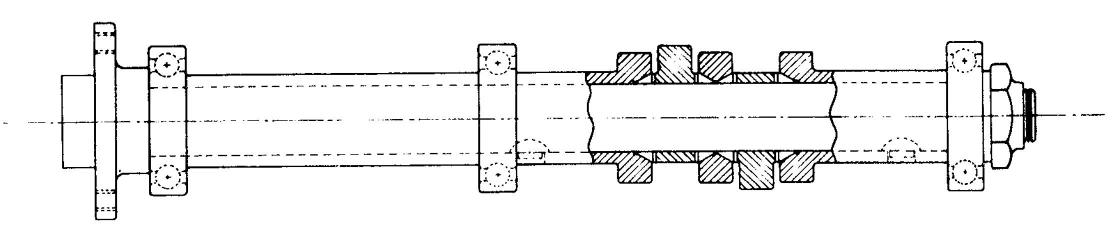

Figure 3.- Adjustable camshaft and valve operating mechanism.

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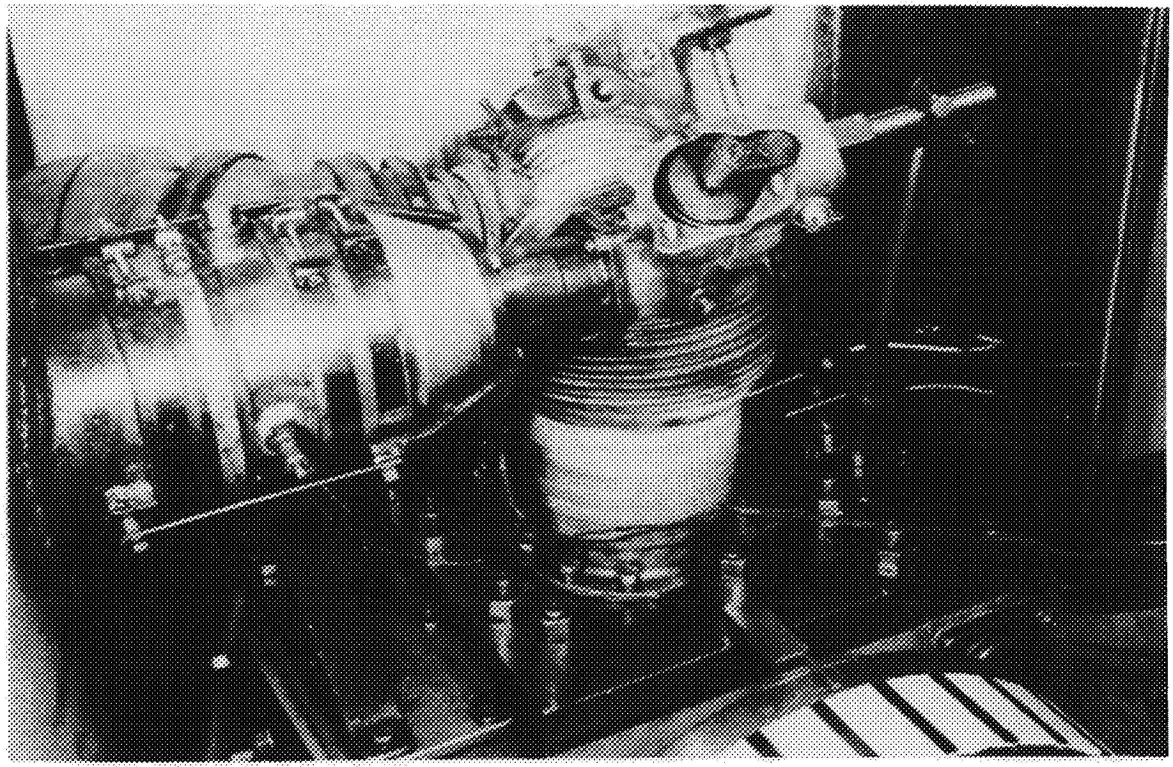

Figure 2.- Engine set-up.

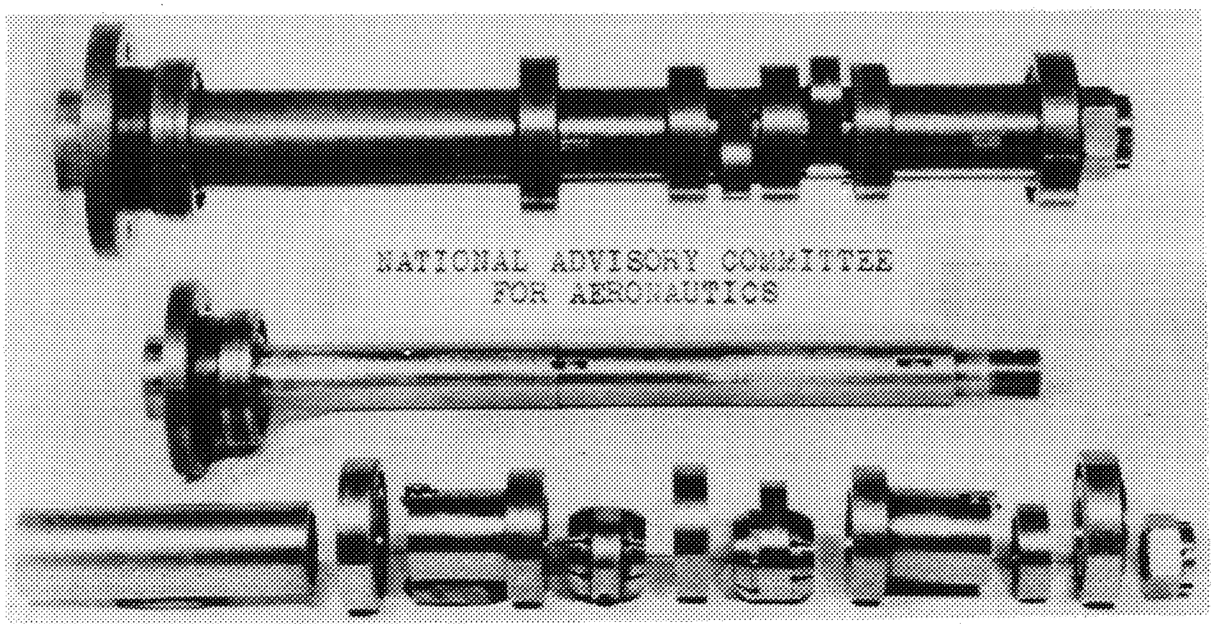

Figure 4.- Special camshaft assembled and disassembled.

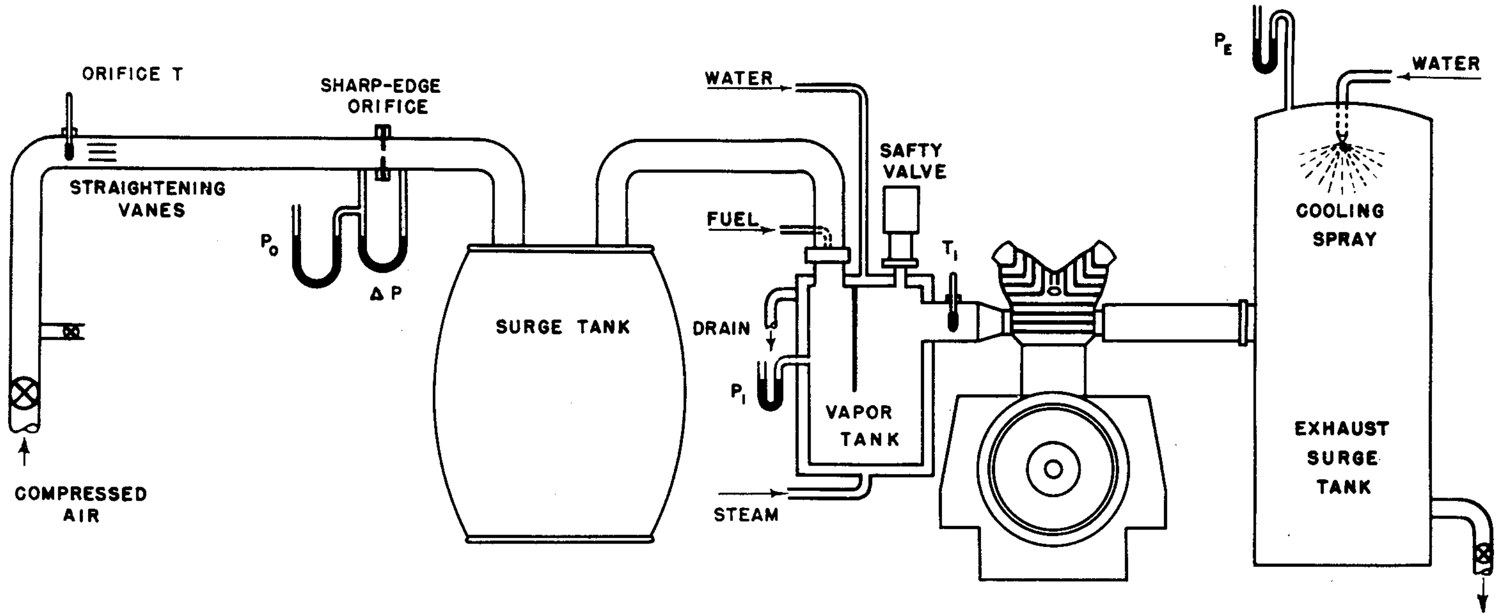

Figure 5.- Schematic diagram of engine setup.

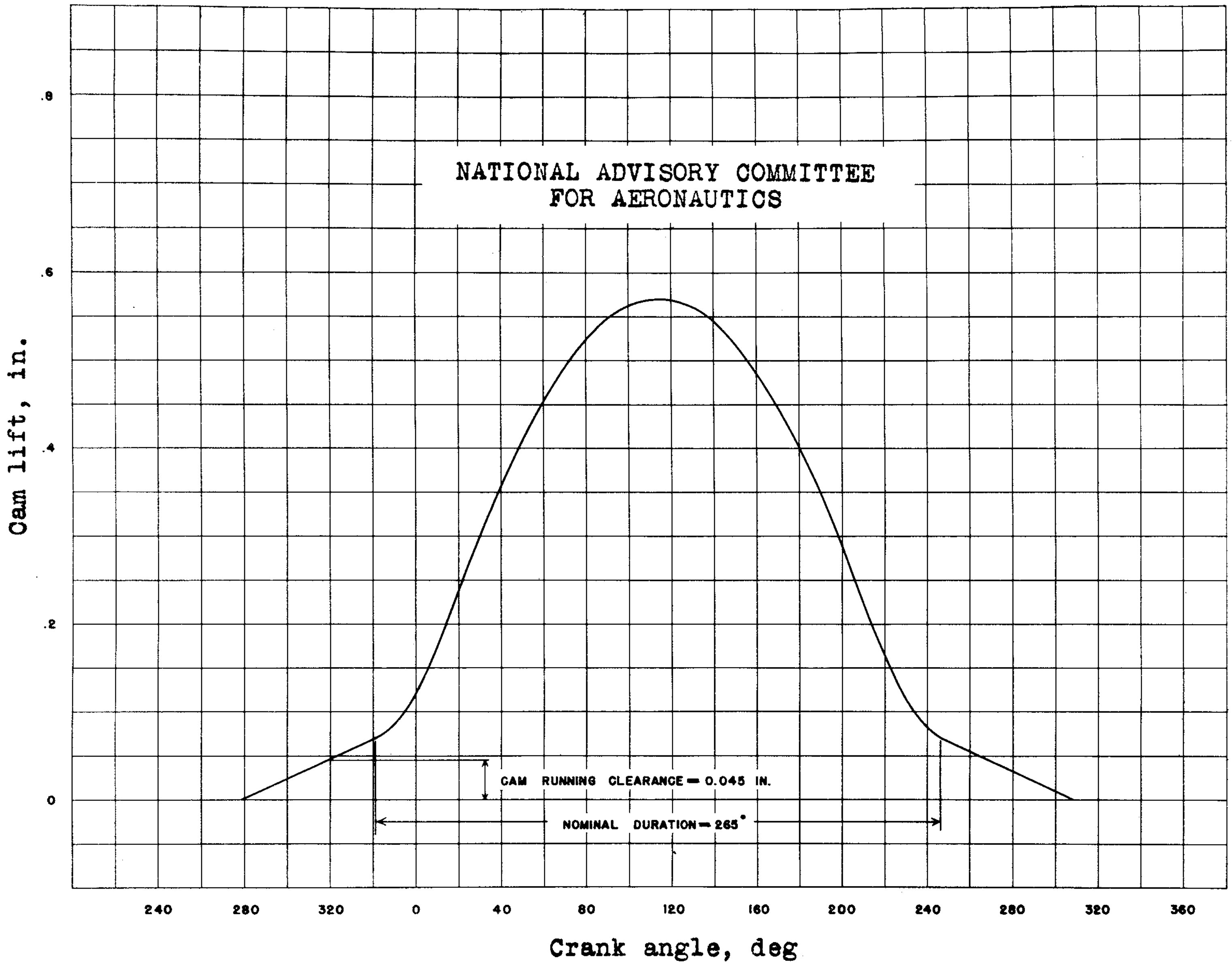

Figure 6.- Typical cam profile.

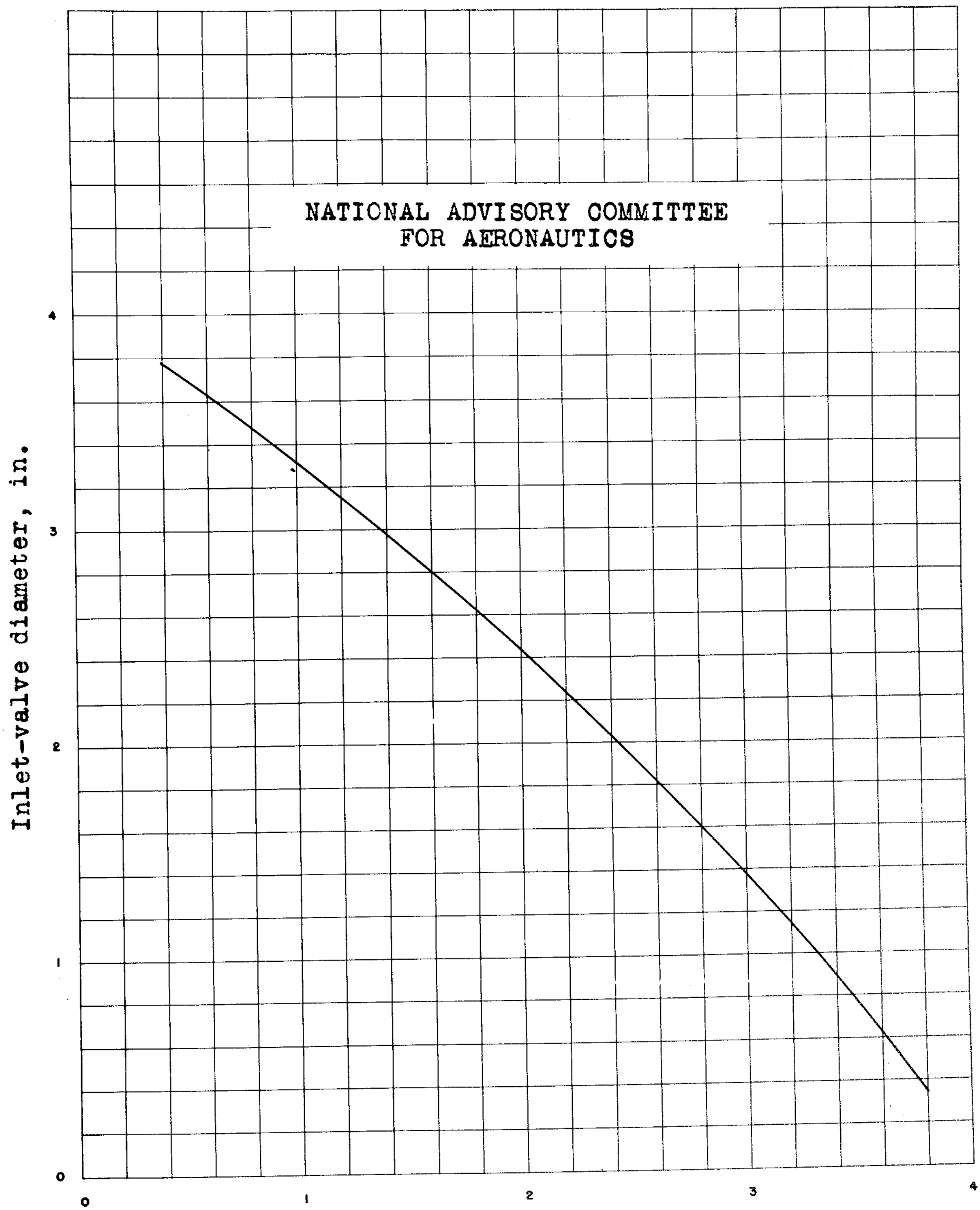

Figure 7.- Valve diameter relationships in a valve-in-head cylinder. (Derived from fig. 1.)

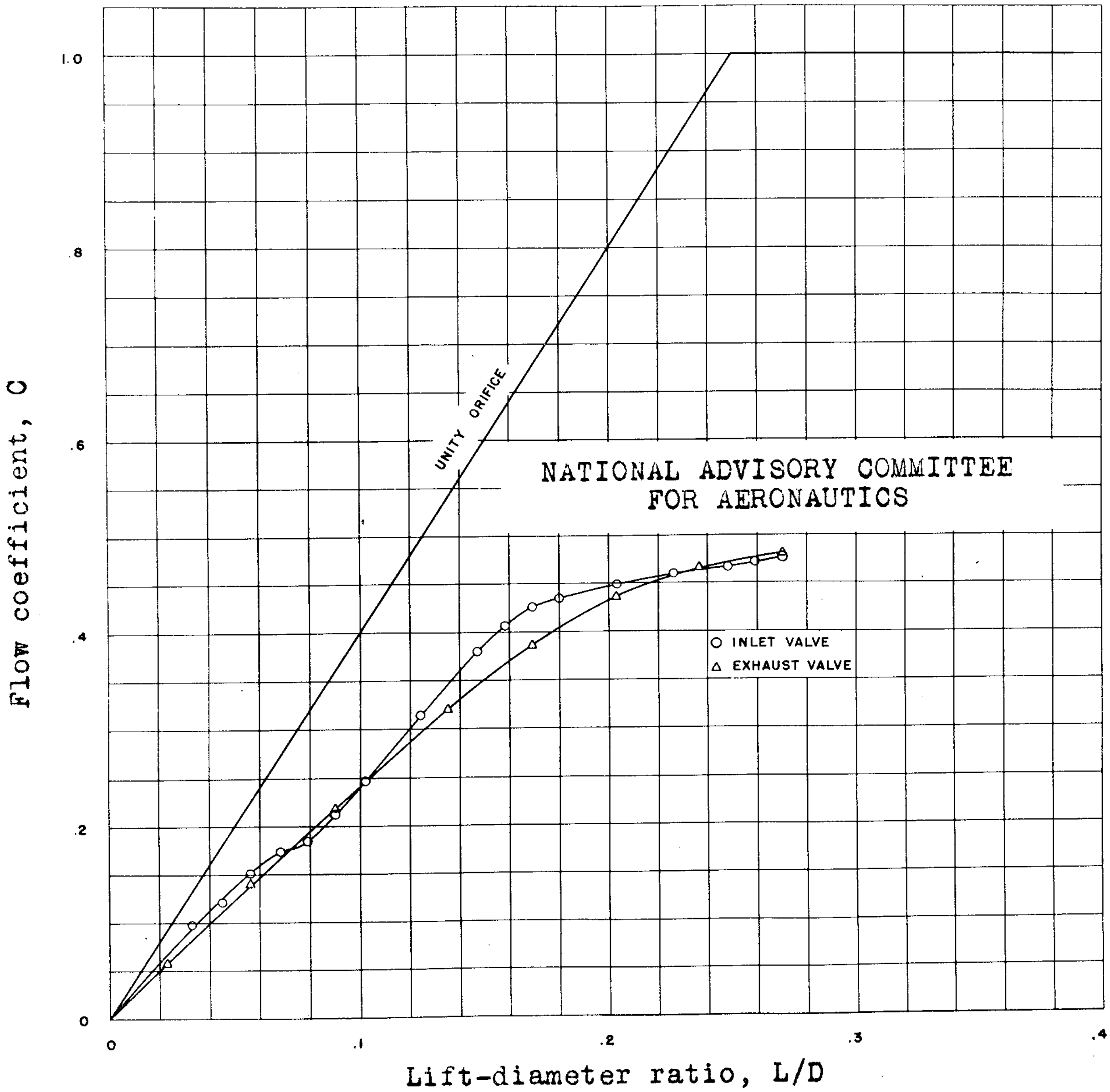

Figure 8.- Valve flow coefficients.

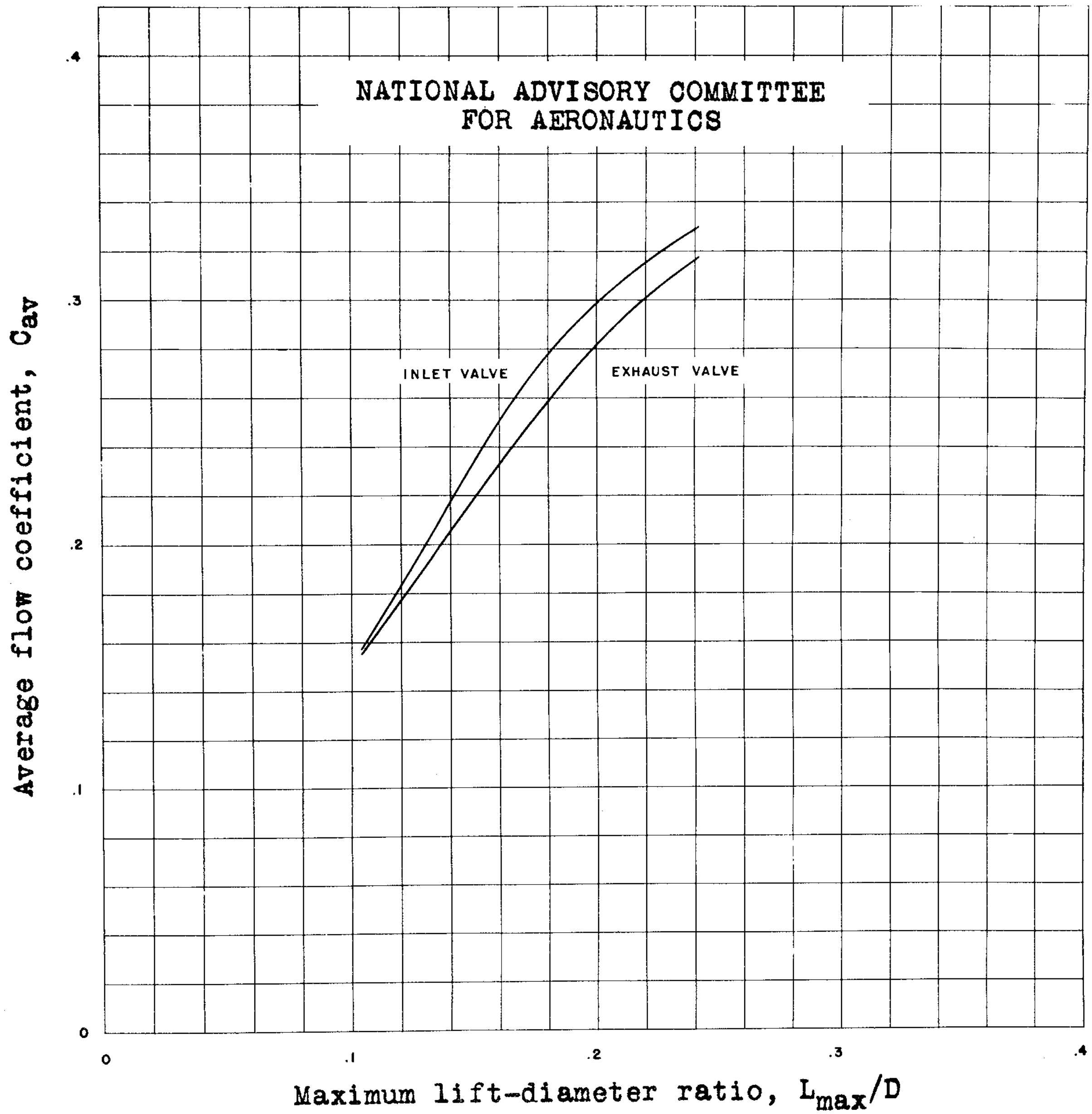

Figure 9.- Average flow coefficients.

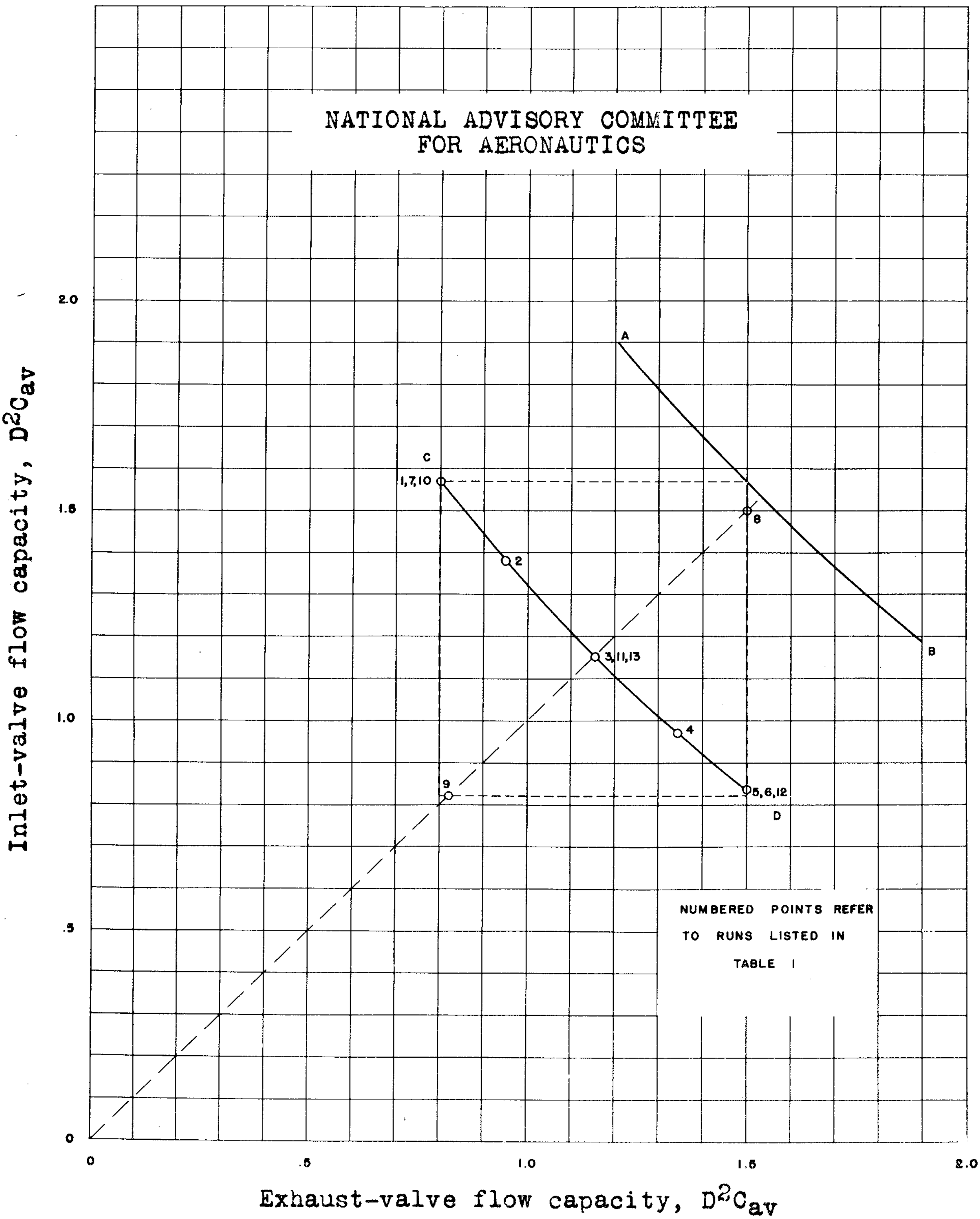

Figure 10.- Valve flow-capacity relationships in a valve-in-head cylinder.

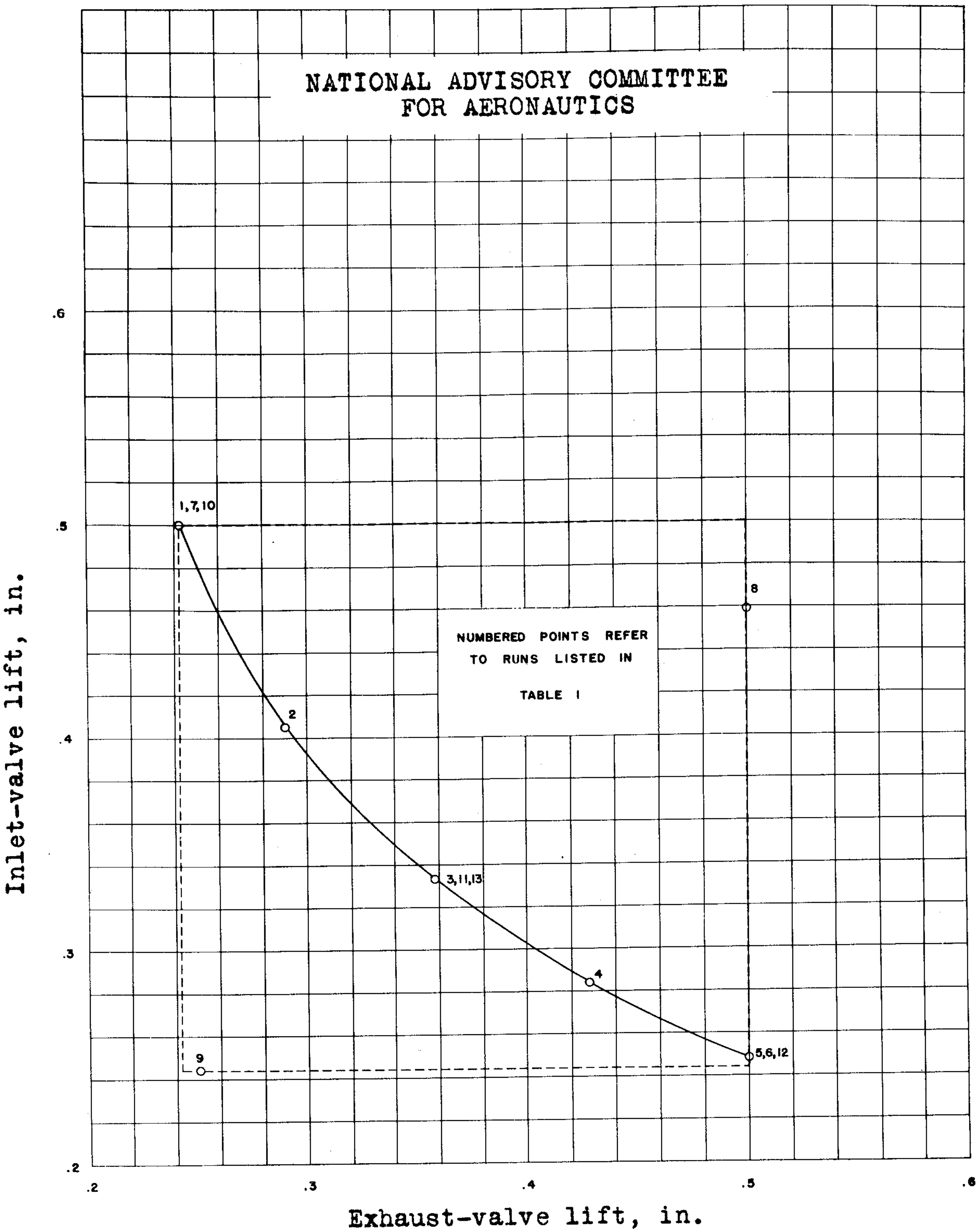

Figure 11.- Valve lifts corresponding to the flow-capacity relationships of figure 10.

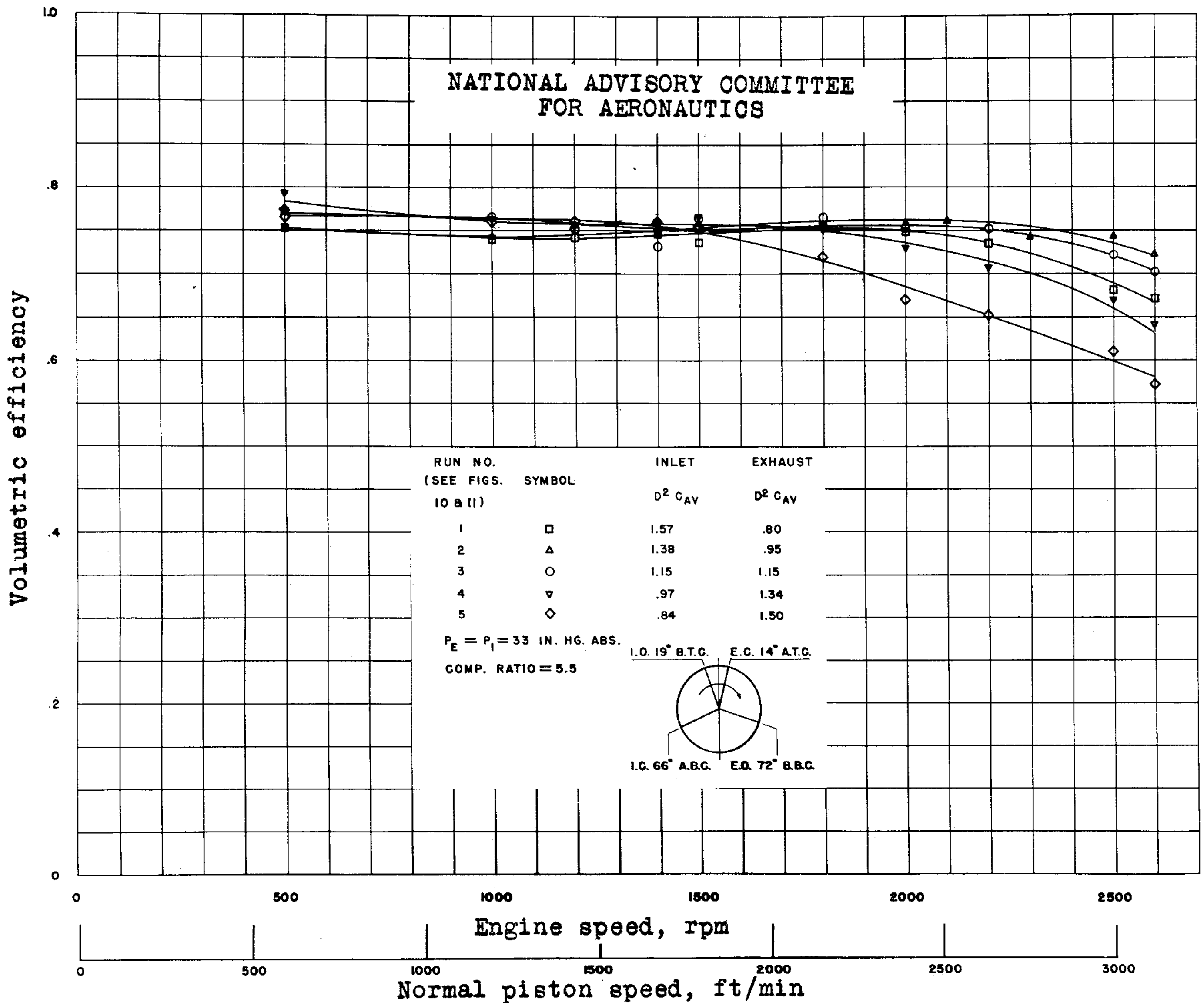

Figure 12.- Volumetric efficiency of an engine with several ratios of exhaust-valve flow capacity to inlet-valve flow capacity.

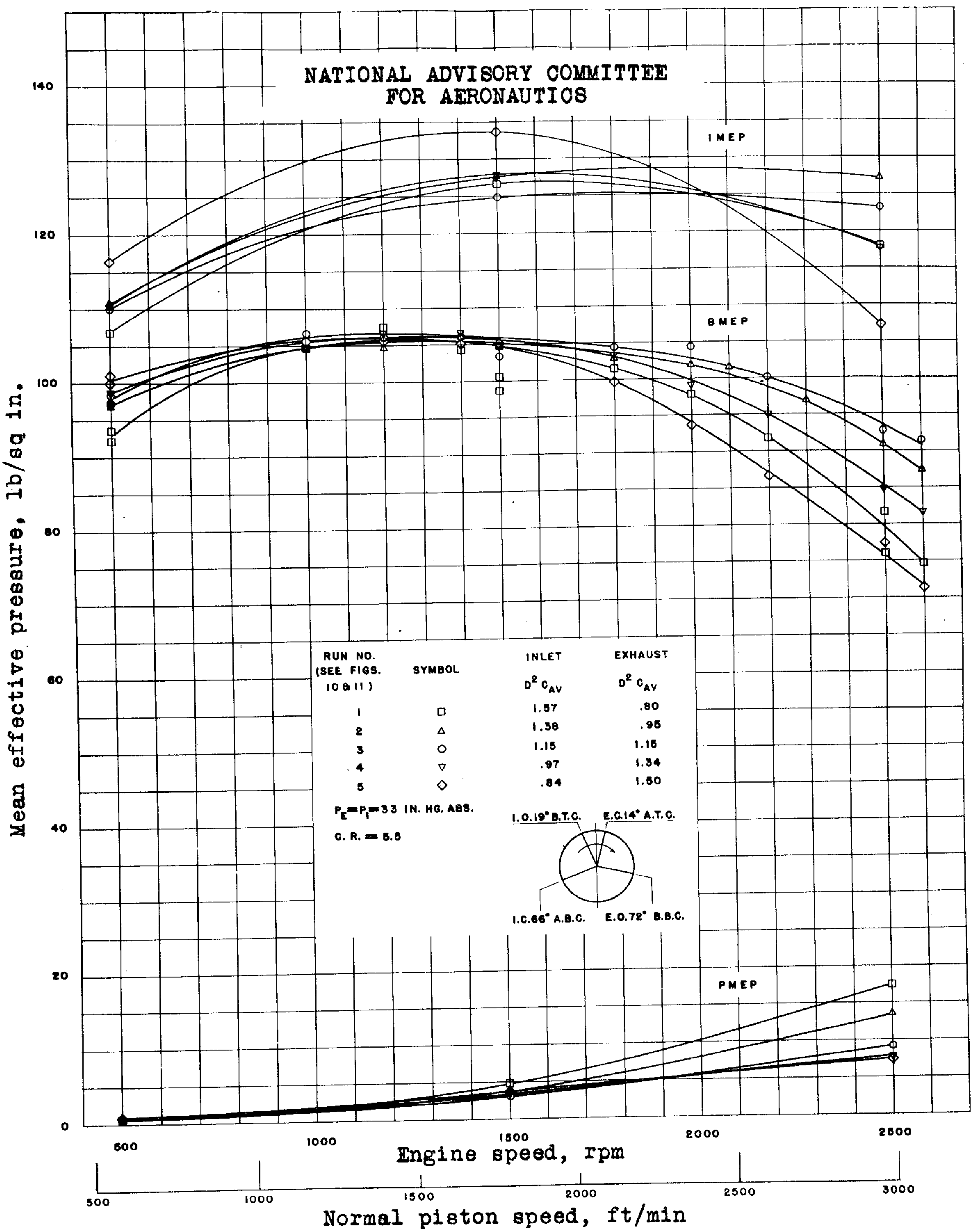

Figure 13.- Performance of an engine with several ratios of exhaust-valve flow capacity to inlet-valve flow capacity.

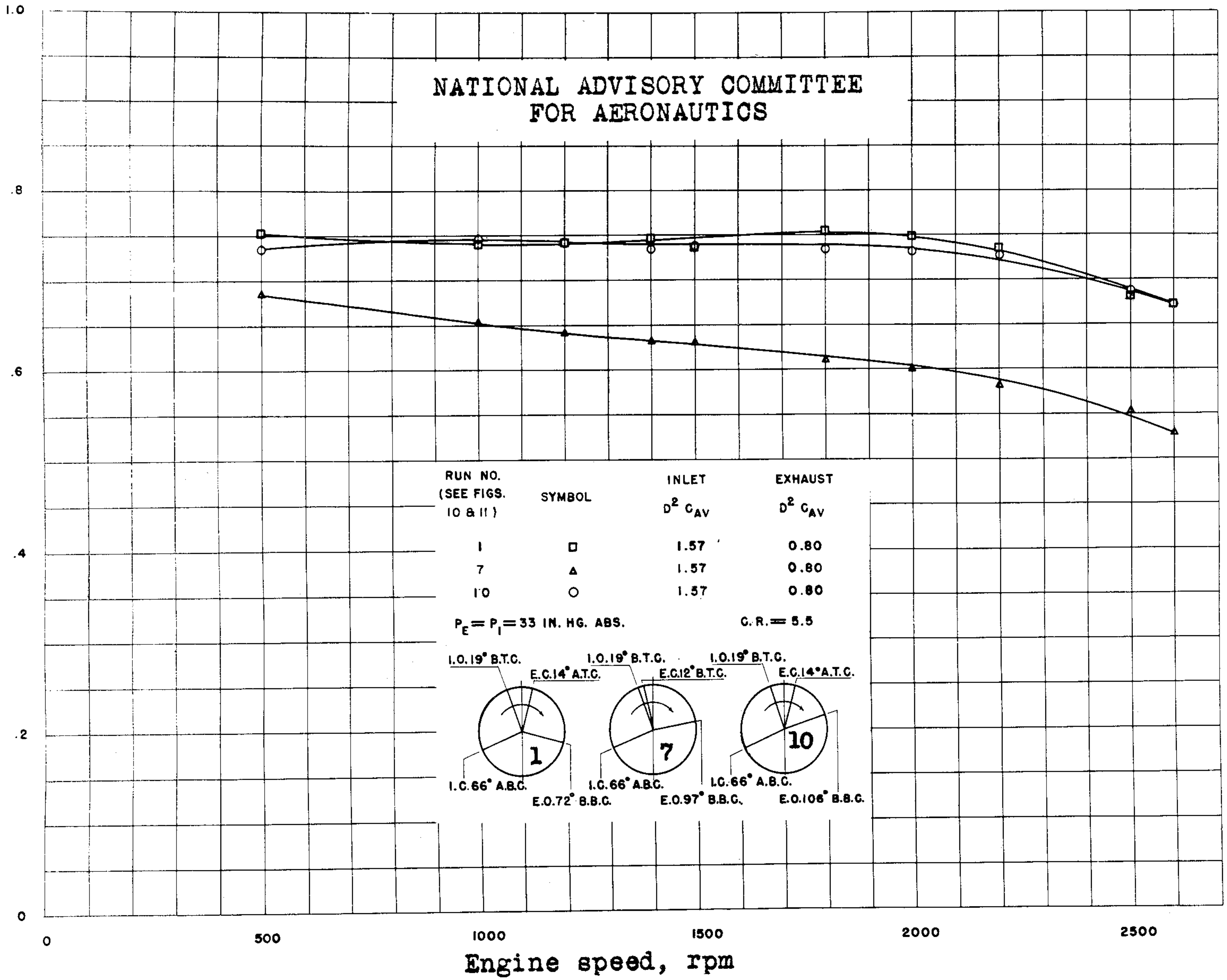

Figure 14.- Volumetric efficiency of an engine with small exhaust-valve flow capacity and three exhaust timing arrangements.

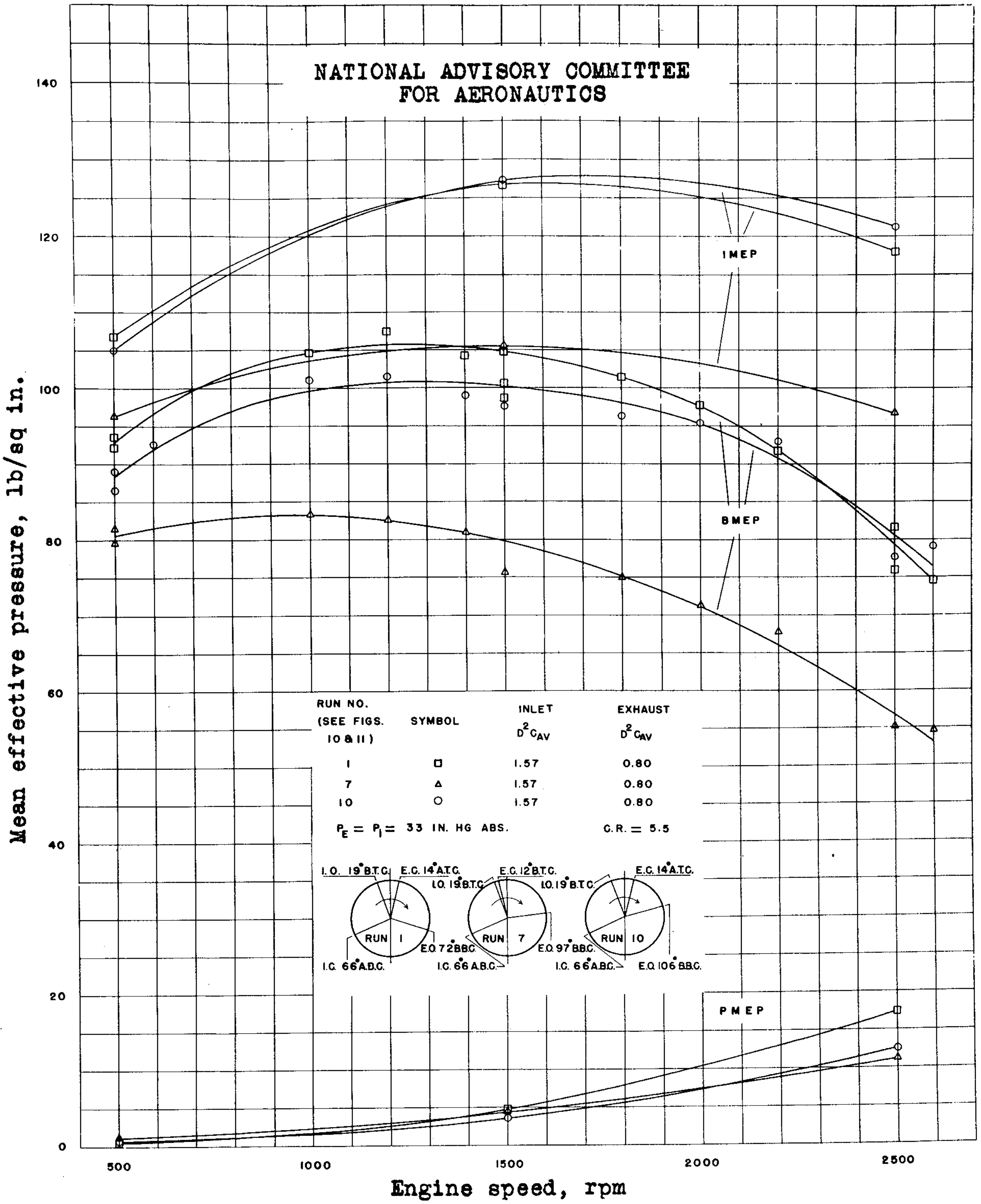

Figure 15.- Performance of an engine with small exhaust-valve flow capacity and three exhaust timing arrangements.

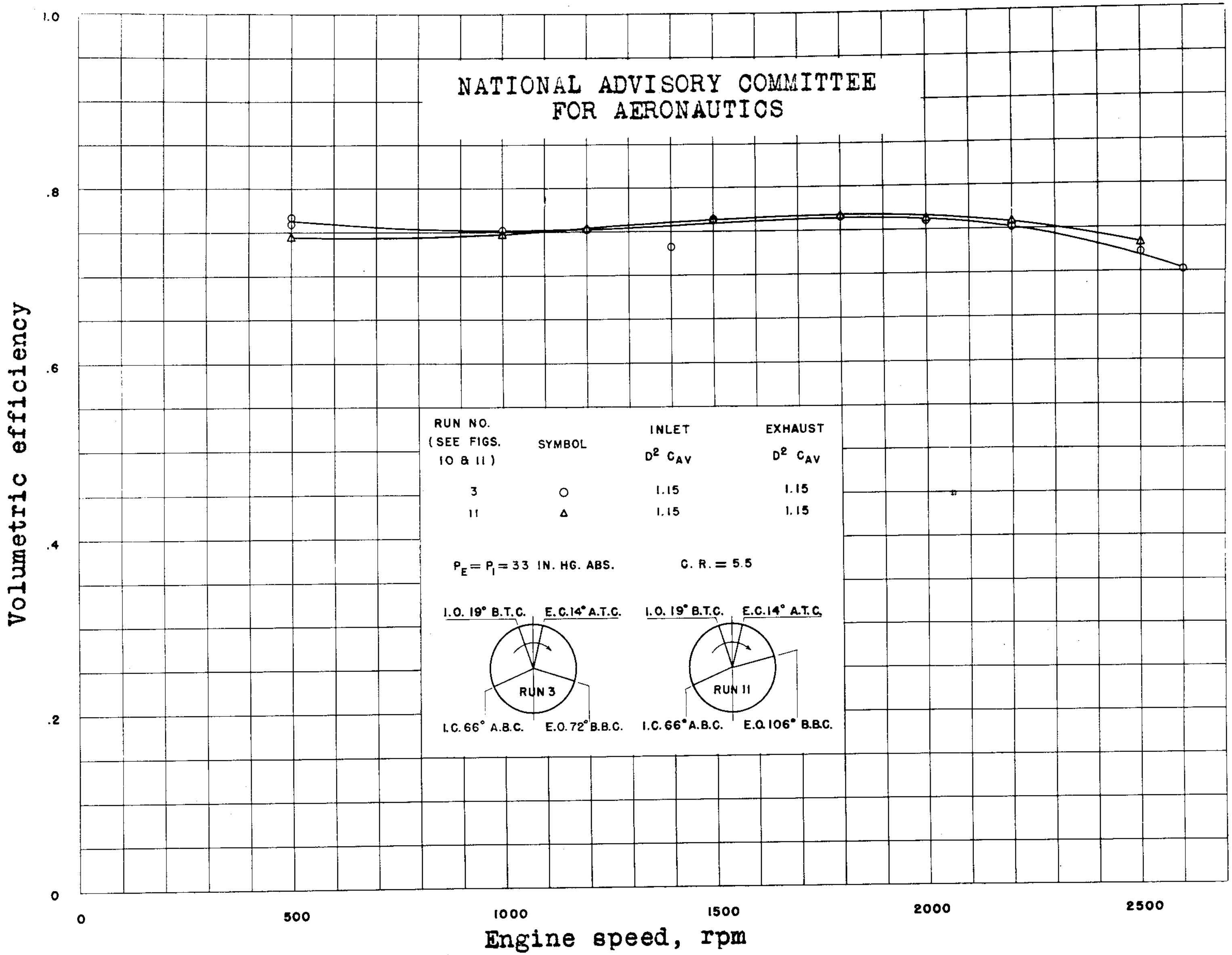

Figure 16.- Volumetric efficiency of an engine with equal inlet and exhaust flow capacities and two exhaust timing arrangements.

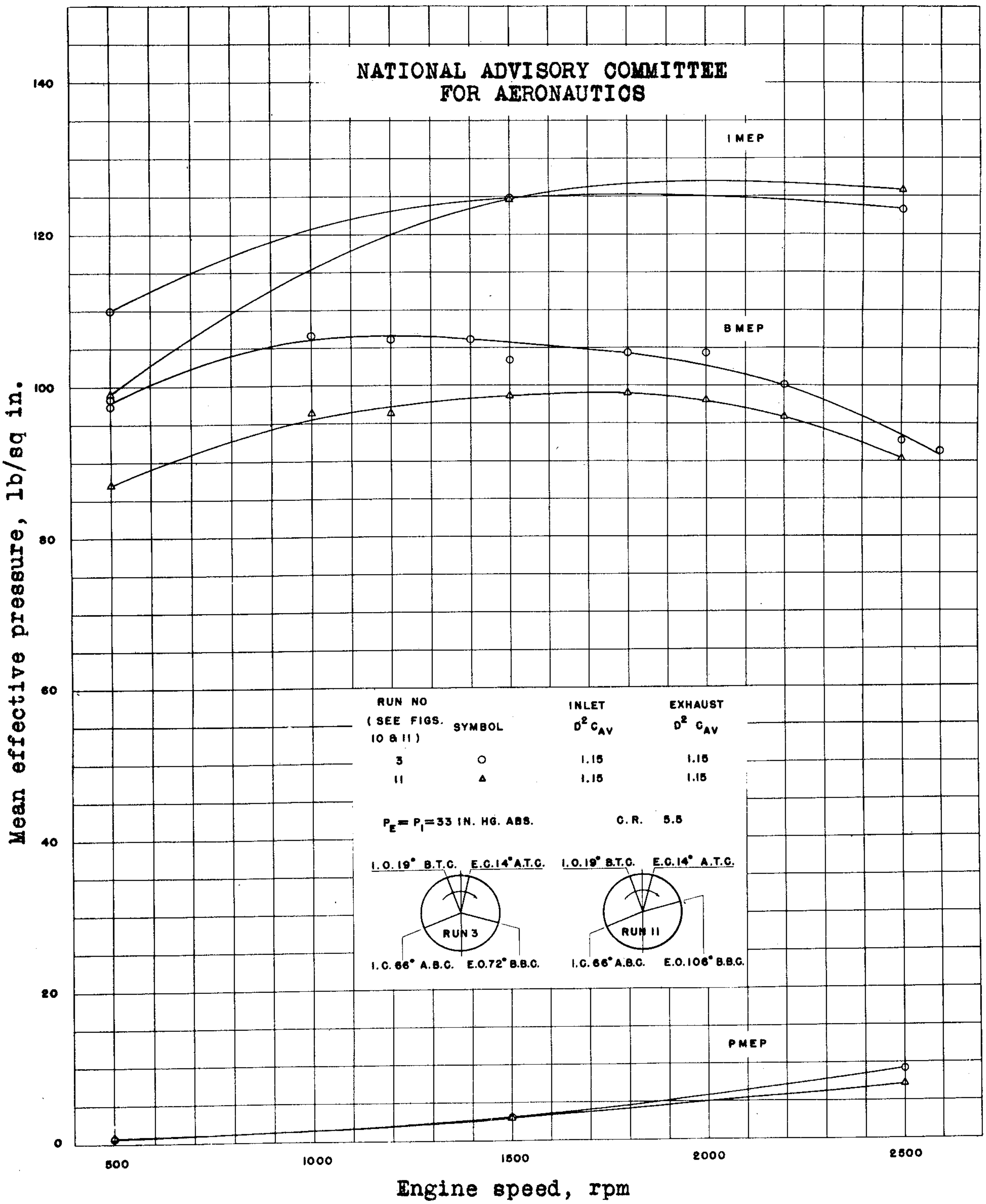

Figure 17.- Performance of an engine with equal inlet and exhaust flow capacities and two exhaust timing arrangements.

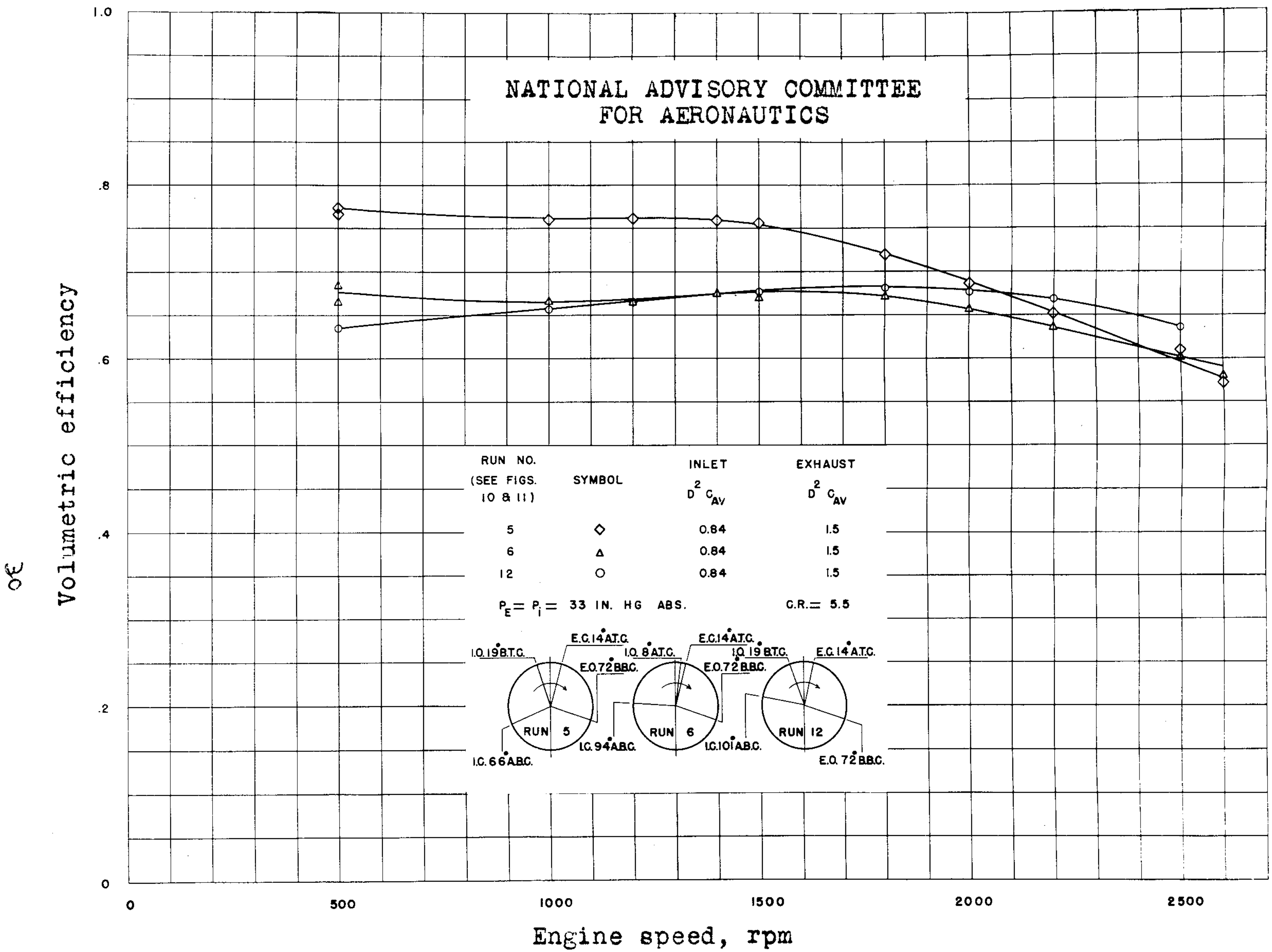

Figure 18.- Volumetric efficiency of an engine with small inlet-valve flow capacity and three inlet timing arrangements.

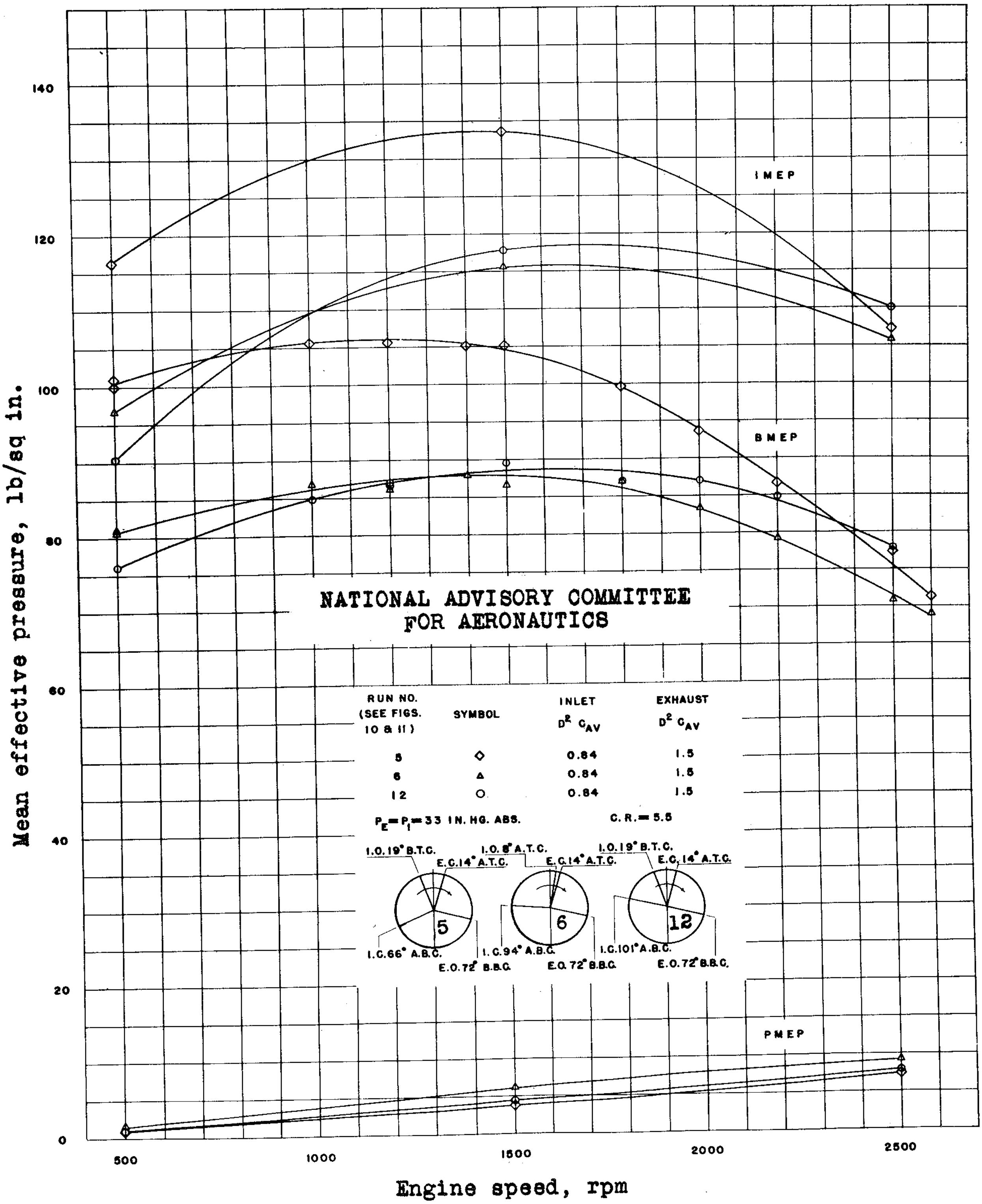

Figure 19.- Performance of an engine with small inlet-valve flow capacity and three inlet timing arrangements.

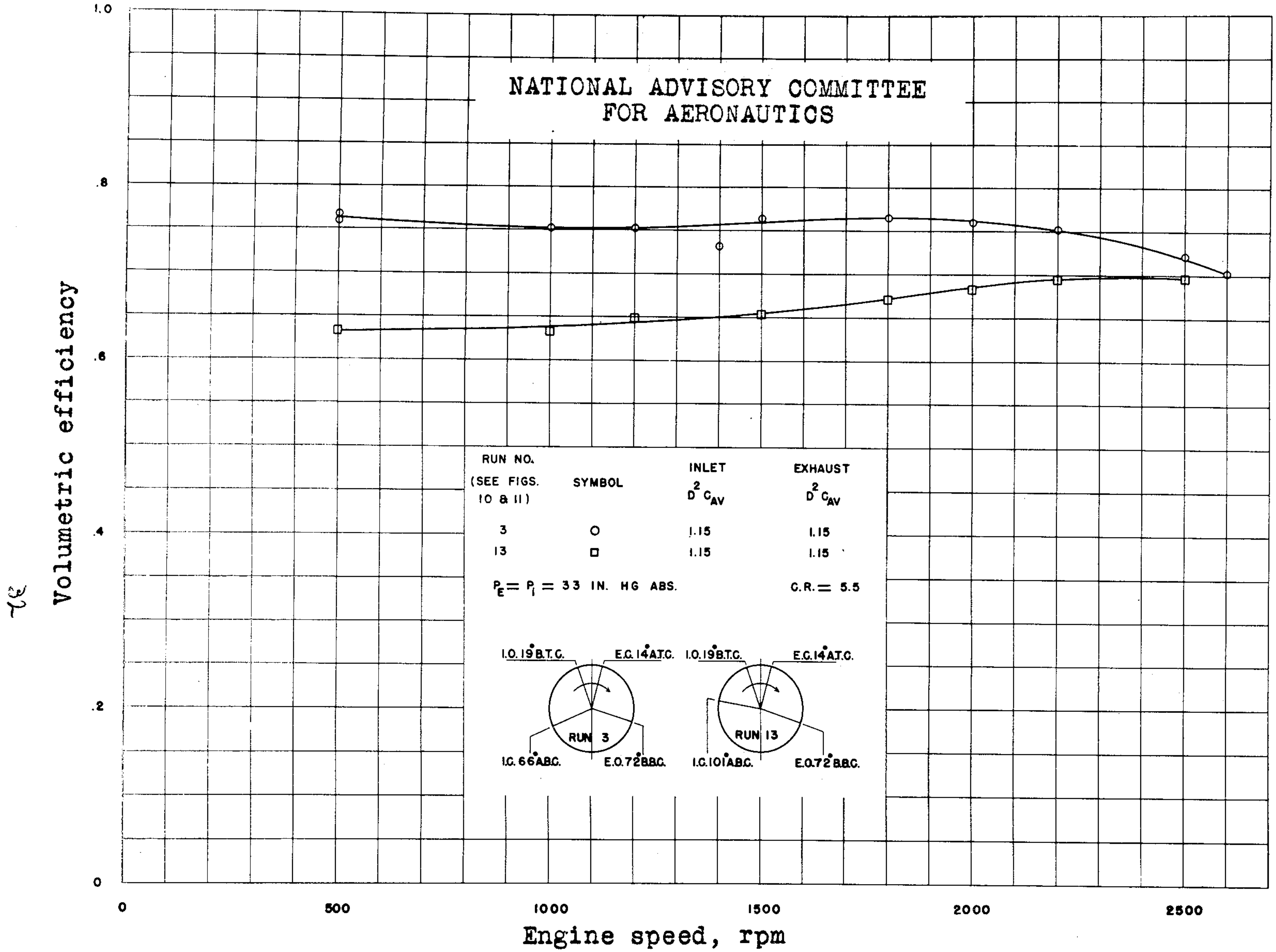

Figure 20.- Volumetric efficiency of an engine with equal inlet and exhaust flow capacities and two inlet timing arrangements.

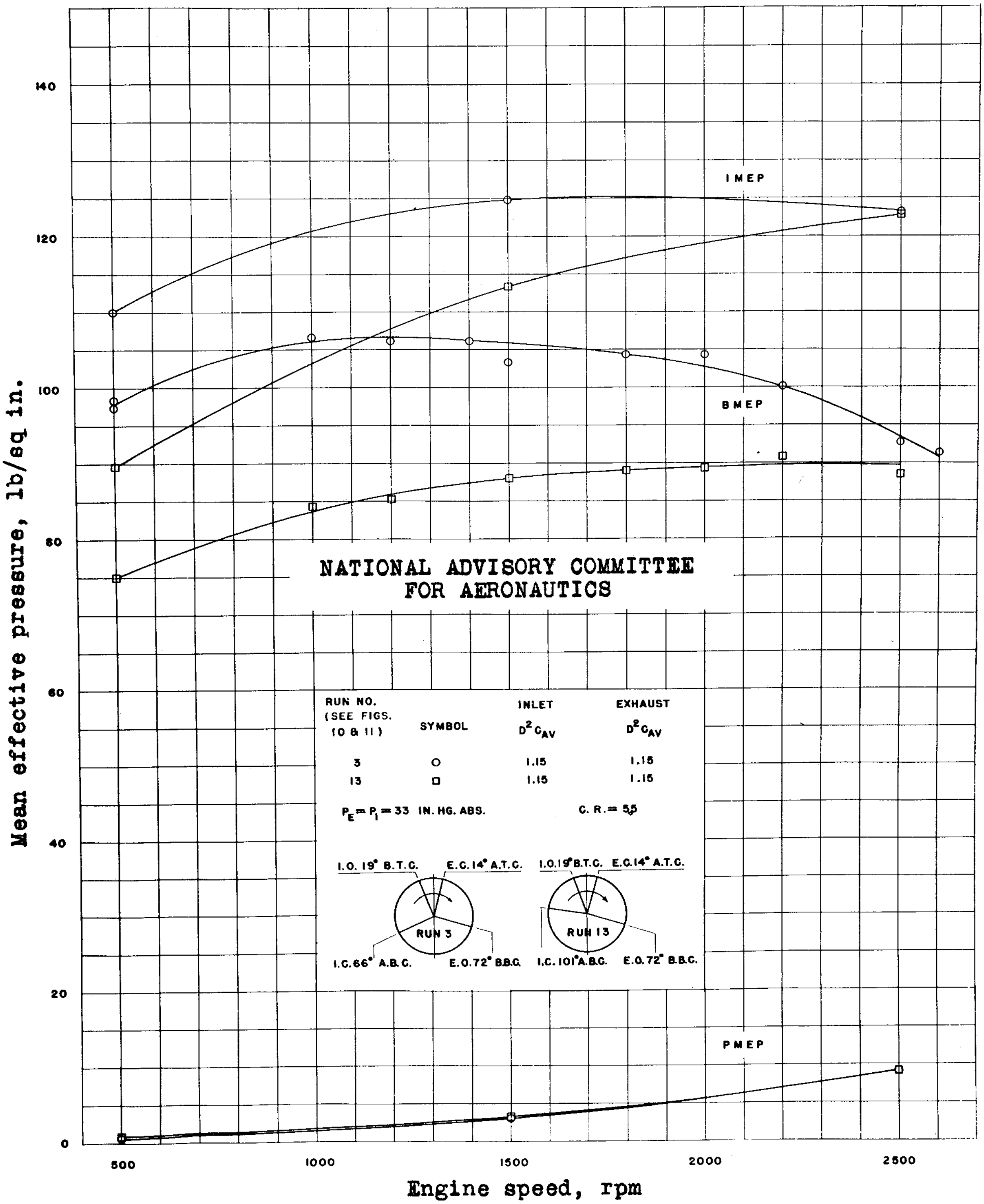

Figure 21.- Performance of an engine with equal inlet- and exhaust-valve flow capacities and two inlet timing arrangements.

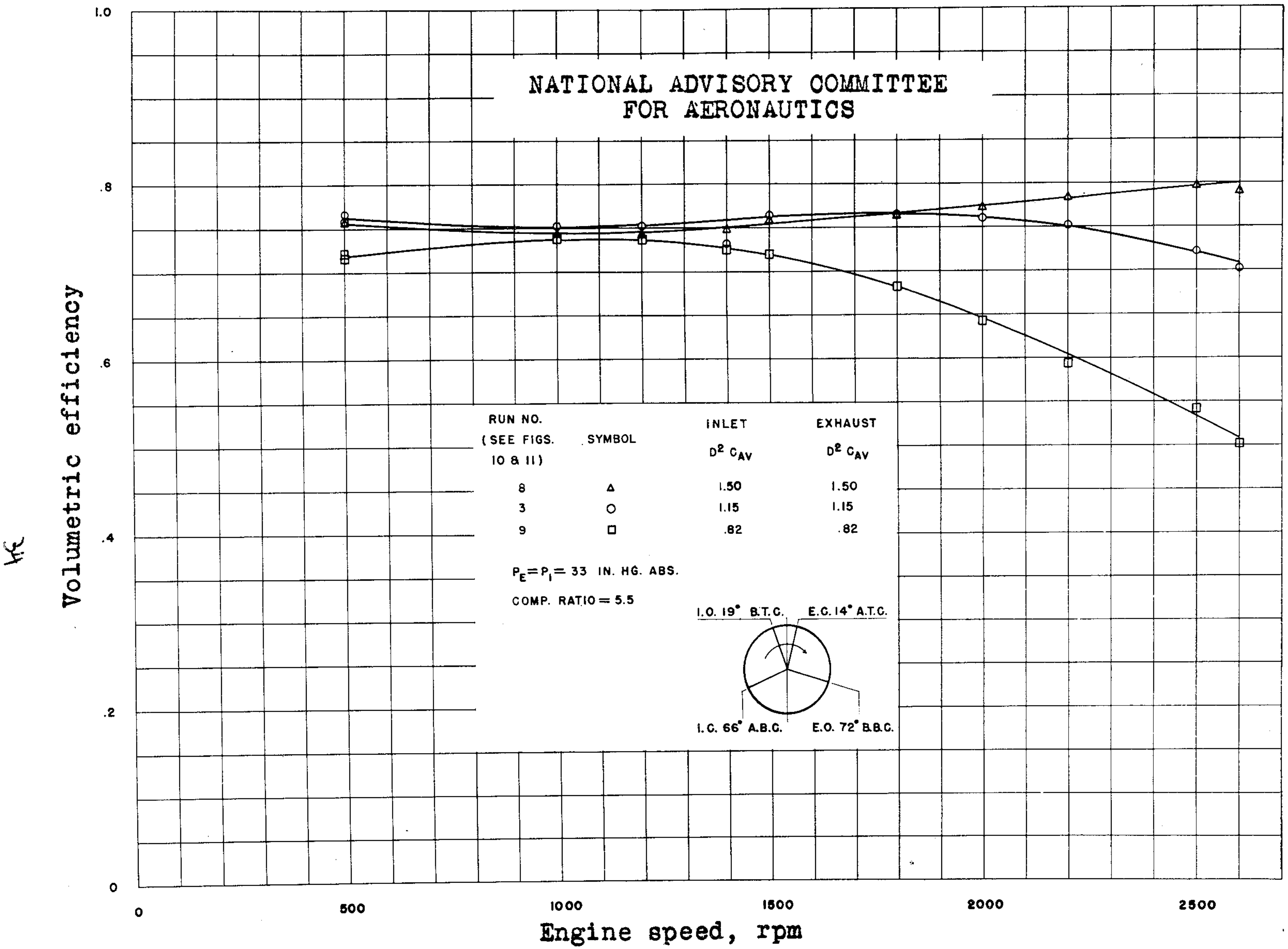

Figure 22.- Volumetric efficiency of an engine with equal inlet- and exhaust-valve flow capacities. Large, intermediate, and small capacities were used.

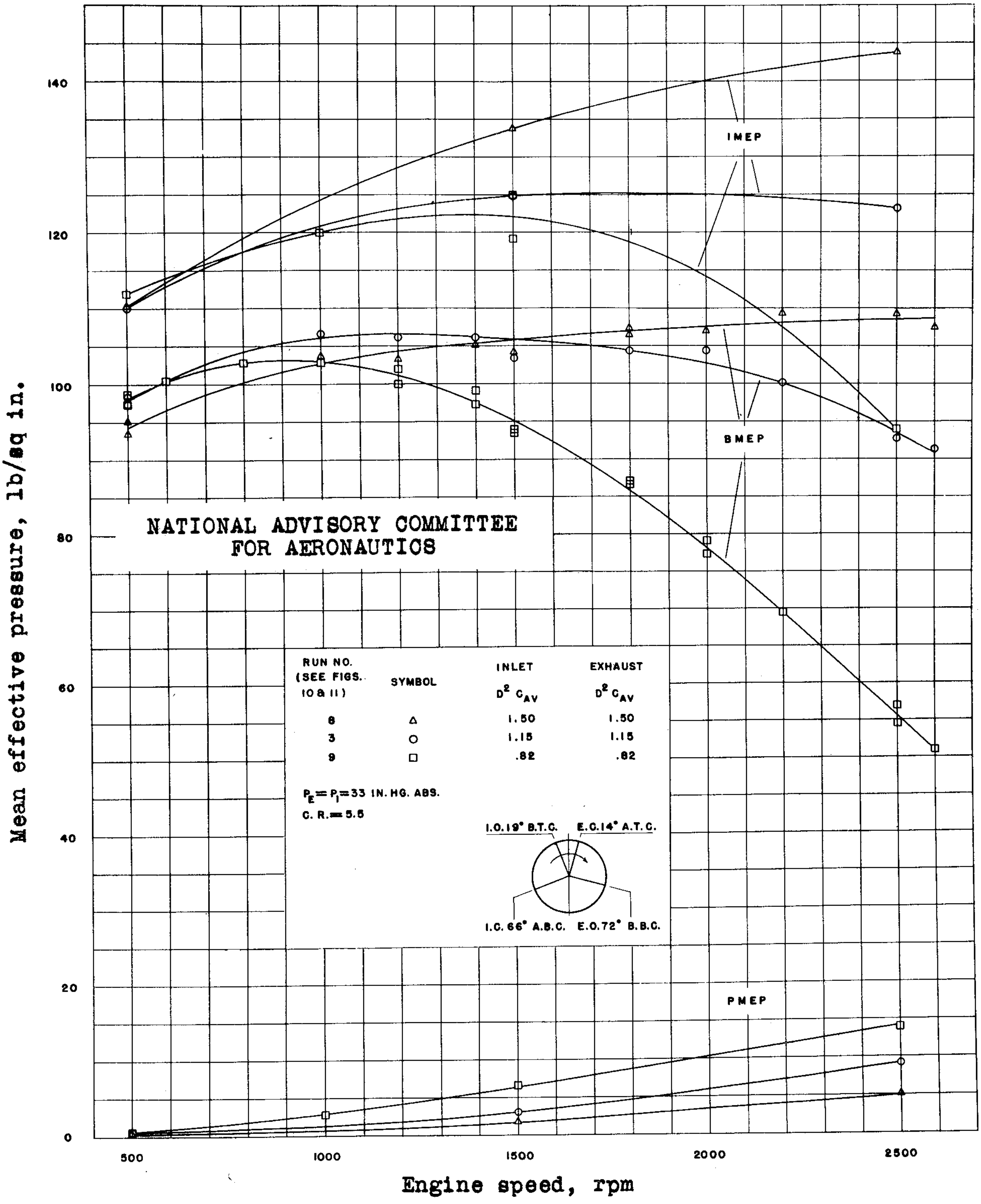

Figure 23.- Performance of an engine with equal inlet- and exhaust-valve flow capacities. Large, intermediate, and small flow capacities were used.

Run	Inlet D^2C_{av}	Exhaust D^2C_{av}	Flow-capacity ratio
1	1.569	0.802	0.51
3	1.151	1.155	1.00
5	0.836	1.500	1.79

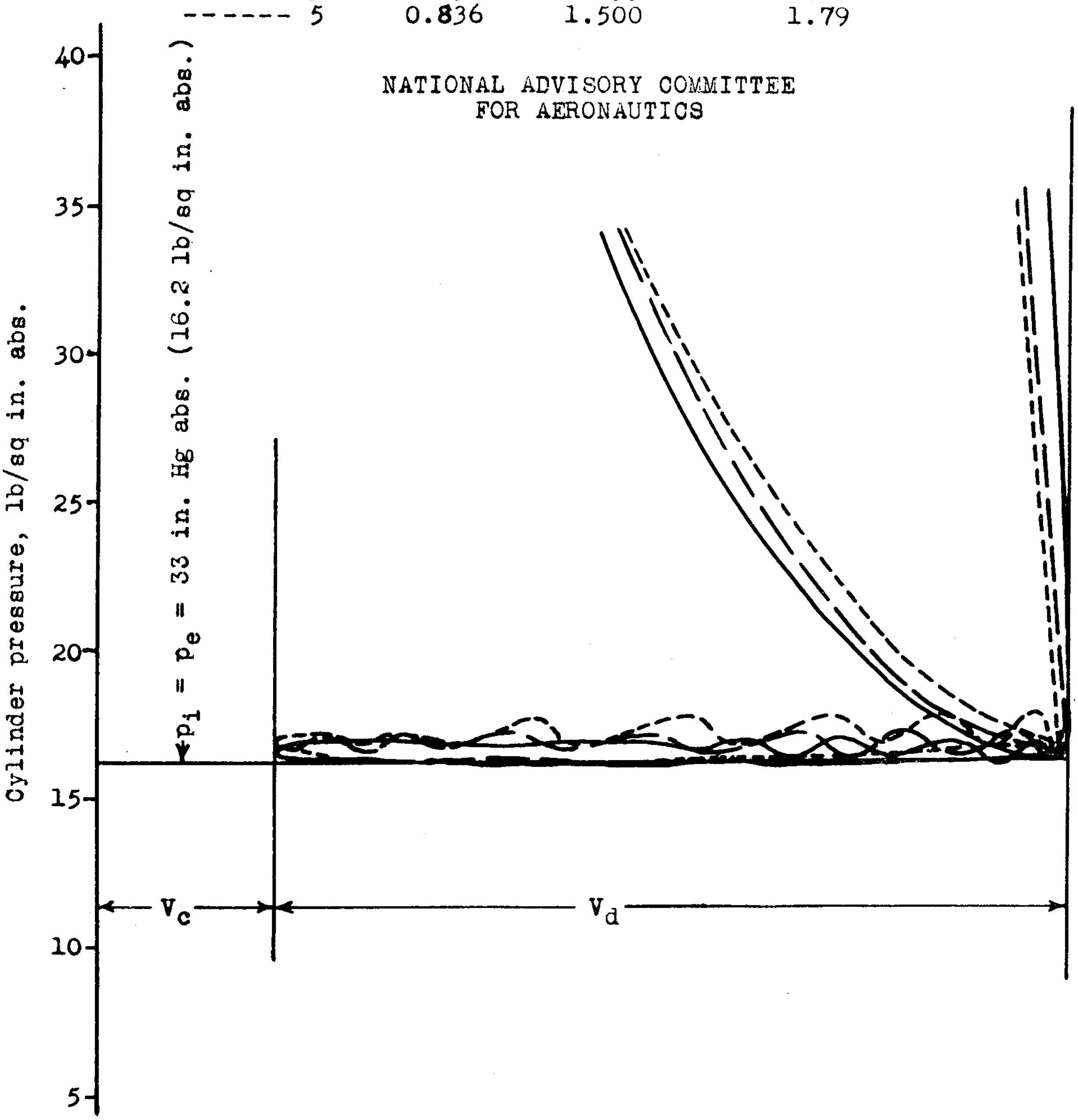

Figure 24.- Light-spring indicator cards. Effect of varying flow-capacity ratio. Engine speed, 500 rpm; normal valve timing (see table I). V_c, clearance volume; V_d, displacement volume.

	Run	Inlet D^2C_{av}	Exhaust D^2C_{av}	Flow-capacity ratio
———	1	1.569	0.802	0.51
— —	3	1.151	1.155	1.00
- - - -	5	0.836	1.500	1.79

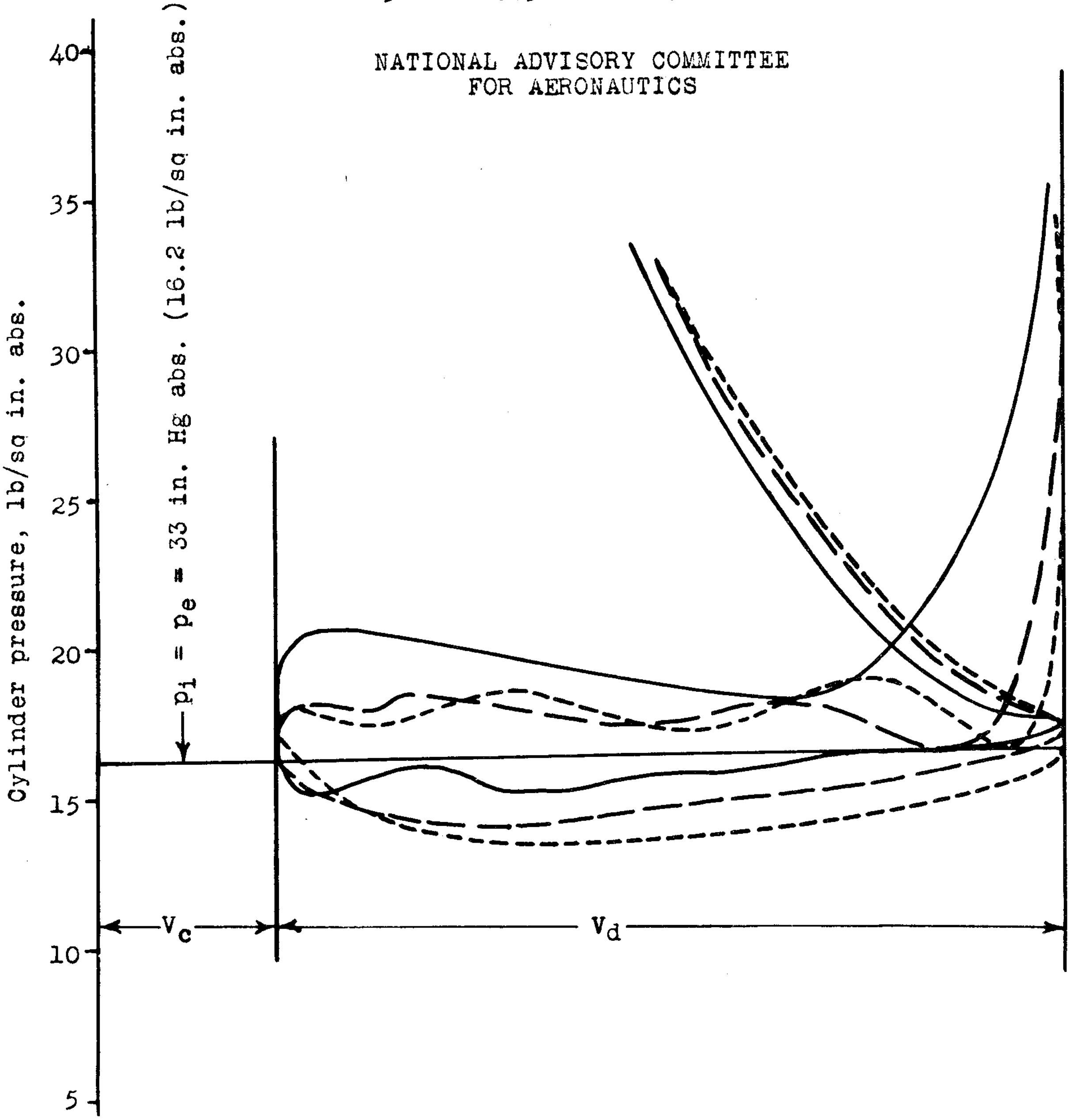

Figure 25.- Light-spring indicator cards. Effect of varying flow-capacity ratio. Engine speed, 1500 rpm; normal valve timing (see table I). V_c, clearance volume; V_d, displacement volume.

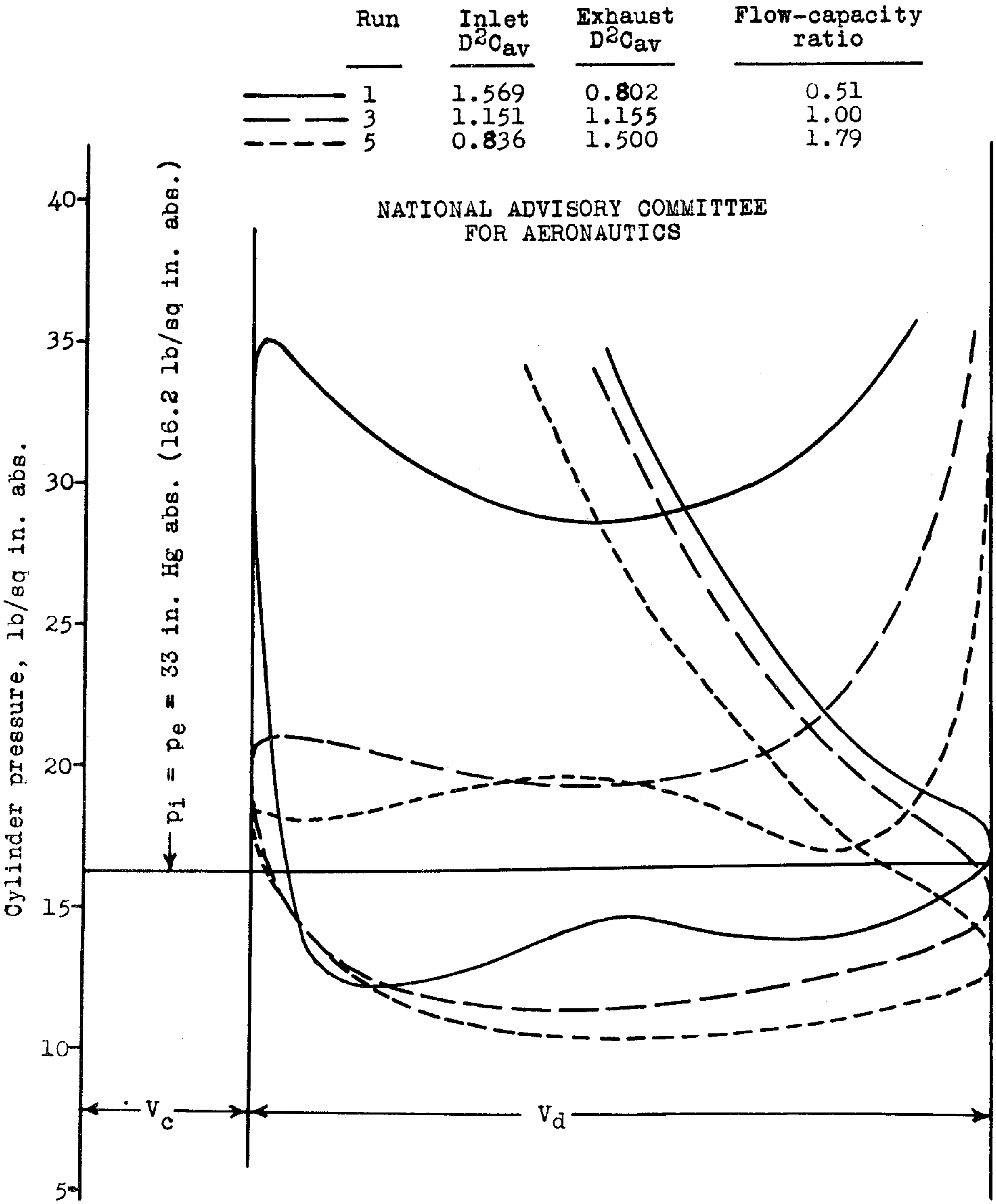

Figure 26.- Light-spring indicator cards. Effect of varying flow-capacity ratio. Engine speed, 2500 rpm; normal valve timing (see table I). V_c, clearance volume; V_d, displacement volume.

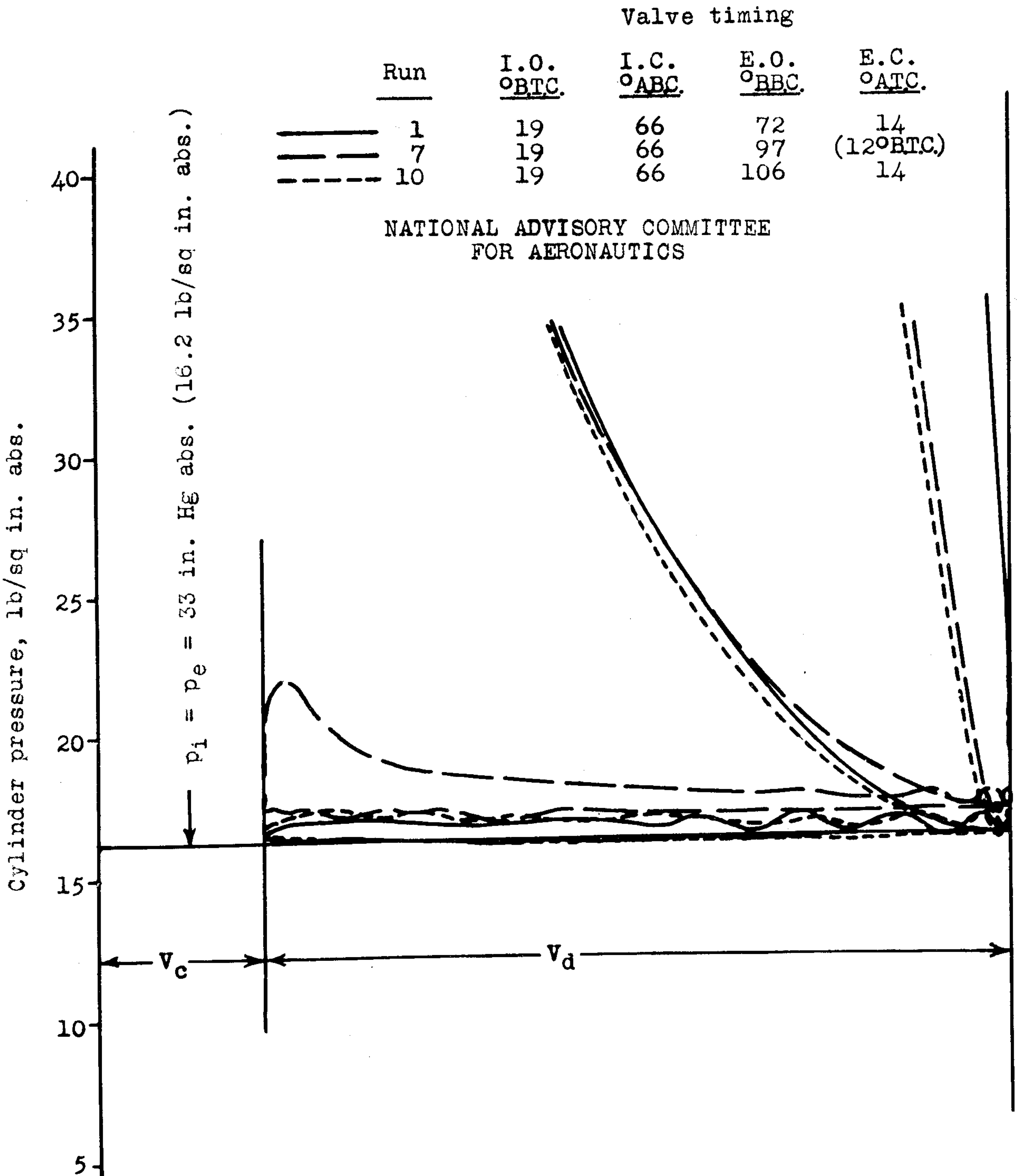

Figure 27.- Light-spring indicator cards. Effect of changing valve timing. Engine speed, 500 rpm; inlet D^2C_{av}, 1.569; exhaust D^2C_{av}, 0.802; flow-capacity ratio, 0.51.

Valve timing

Run	I.O. °BTC	I.C. °ABC	E.O. °BBC	E.C. °ATC
1	19	66	72	14
7	19	66	97	(12°BTC)
10	19	66	106	14

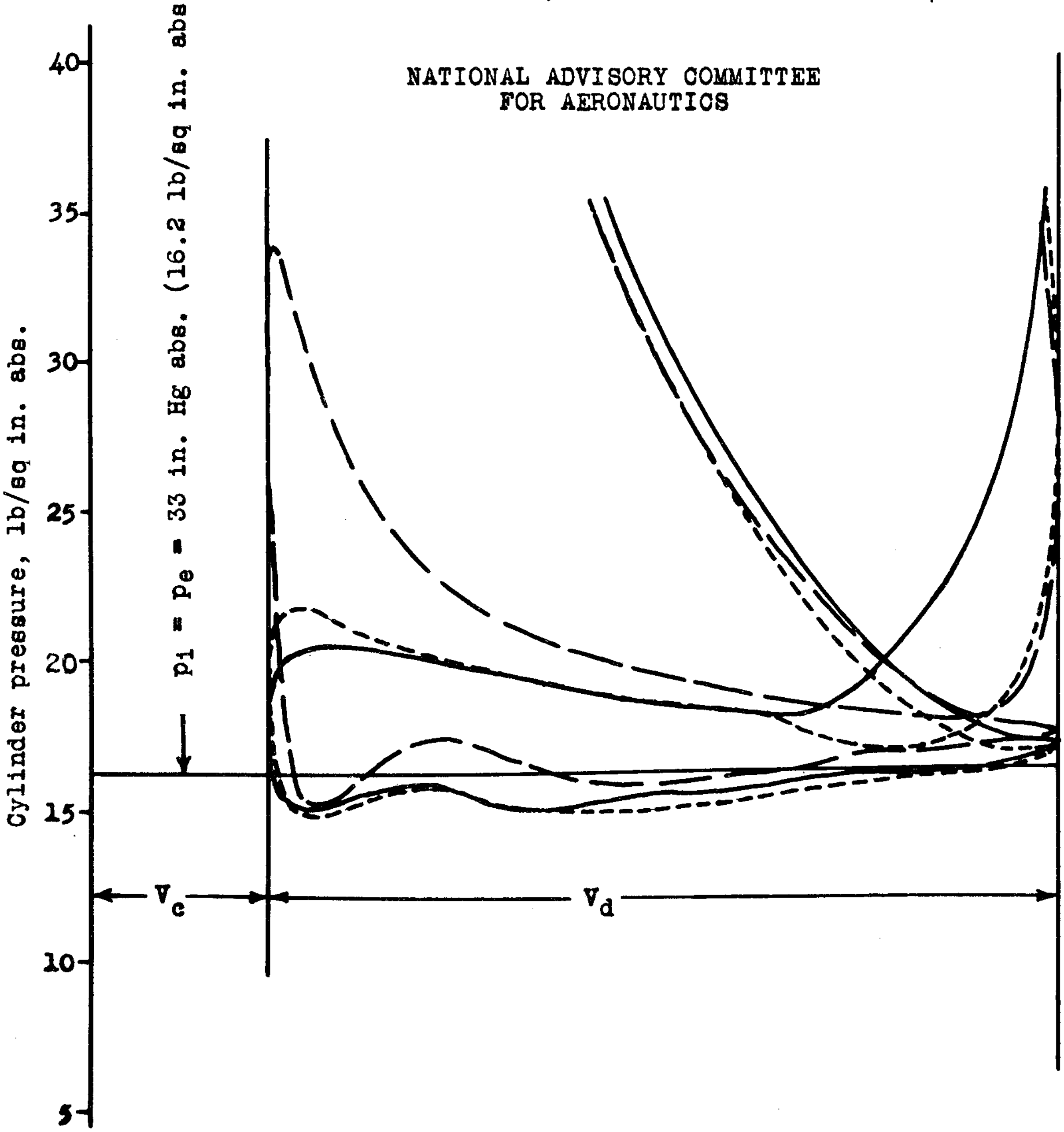

Figure 28.- Light-spring indicator cards. Effect of changing valve timing. Engine speed, 1500 rpm; inlet D^2C_{av}, 1.569; exhaust D^2C_{av}, 0.802; flow-capacity ratio, 0.51.

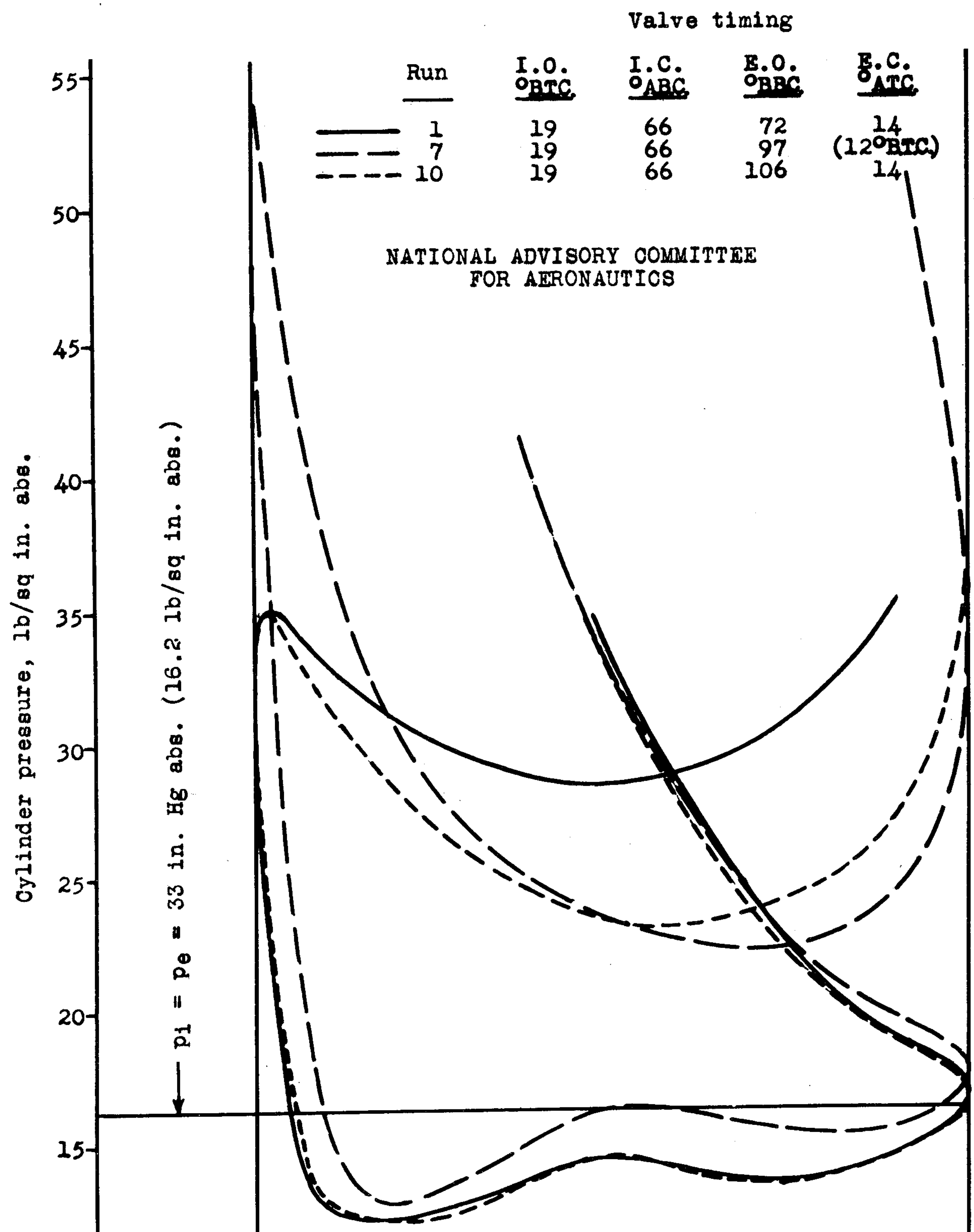

Figure 29.- Light-spring indicator cards. Effect of changing valve timing. Engine speed, 2500 rpm; inlet D^2C_{av}, 1.569; exhaust D^2C_{av}, 0.802; flow-capacity ratio, 0.51.

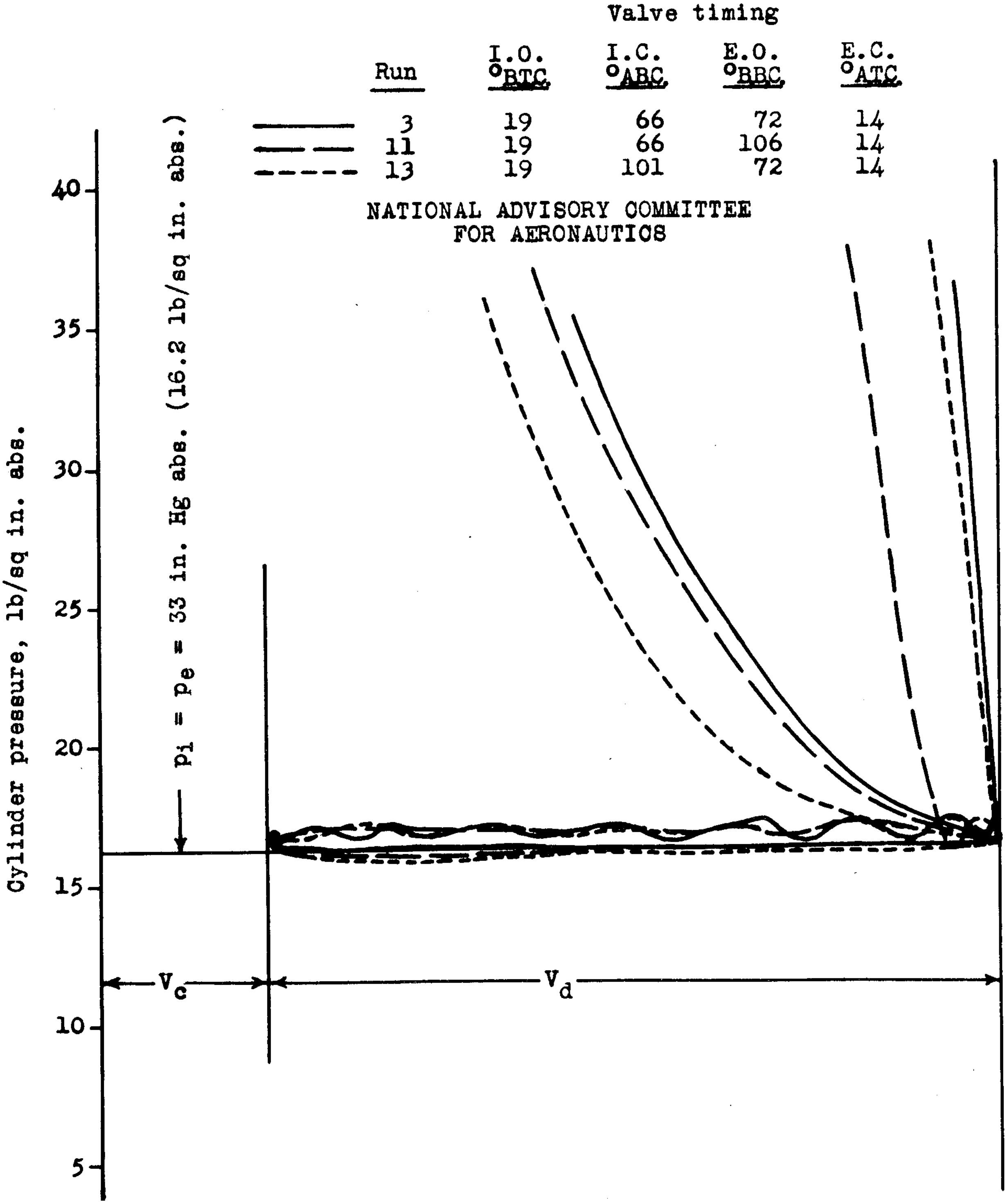

Figure 30.- Light-spring indicator cards. Effect of changing valve timing. Engine speed, 500 rpm; inlet D^2C_{av}, 1.151; exhaust D^2C_{av}, 1.155; flow-capacity ratio, 1.00.

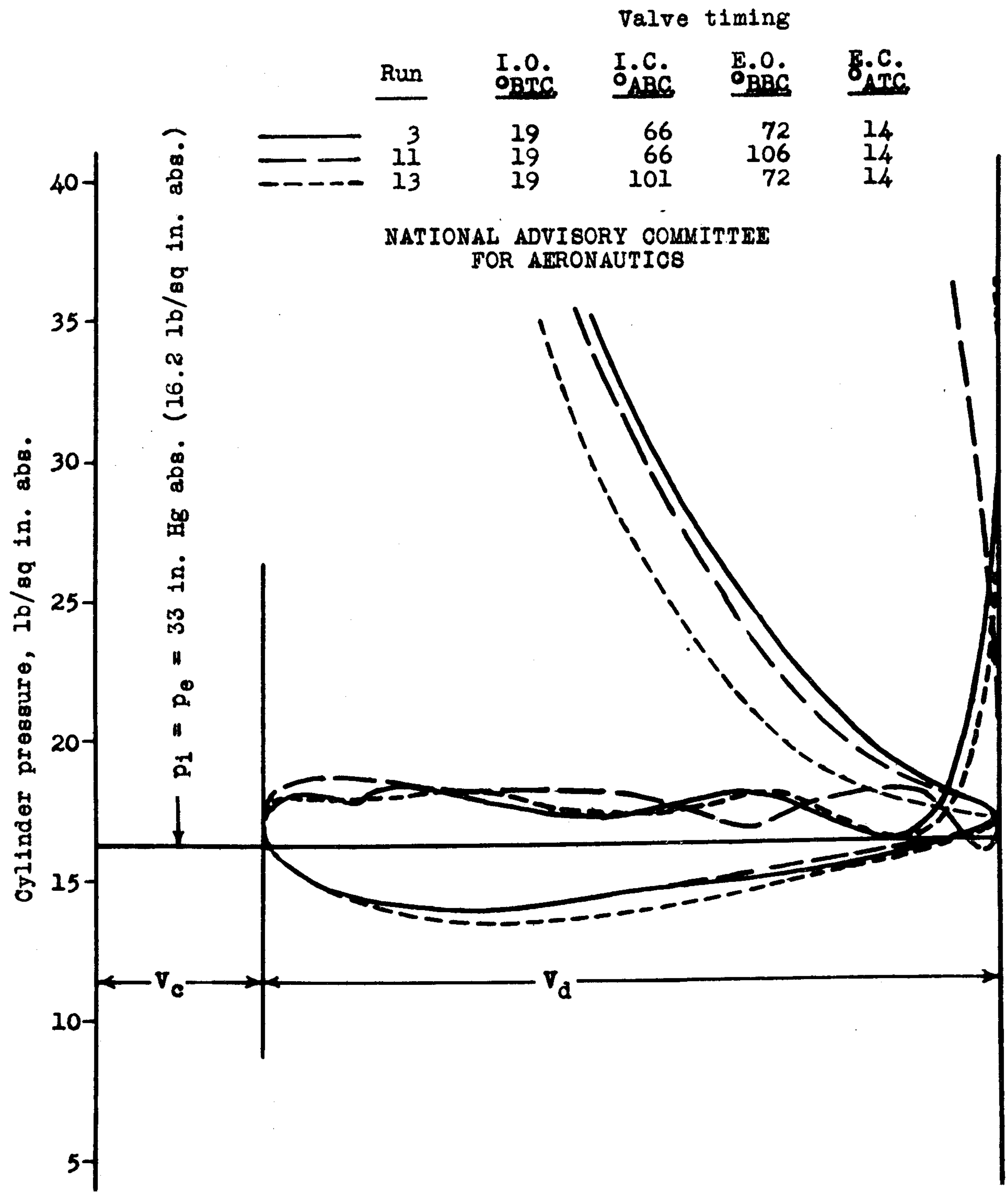

Figure 31.- Light-spring indicator cards. Effect of changing valve timing. Engine speed, 1500 rpm; inlet D^2C_{av}, 1.151; exhaust D^2C_{av}, 1.155; flow-capacity ratio, 1.00.

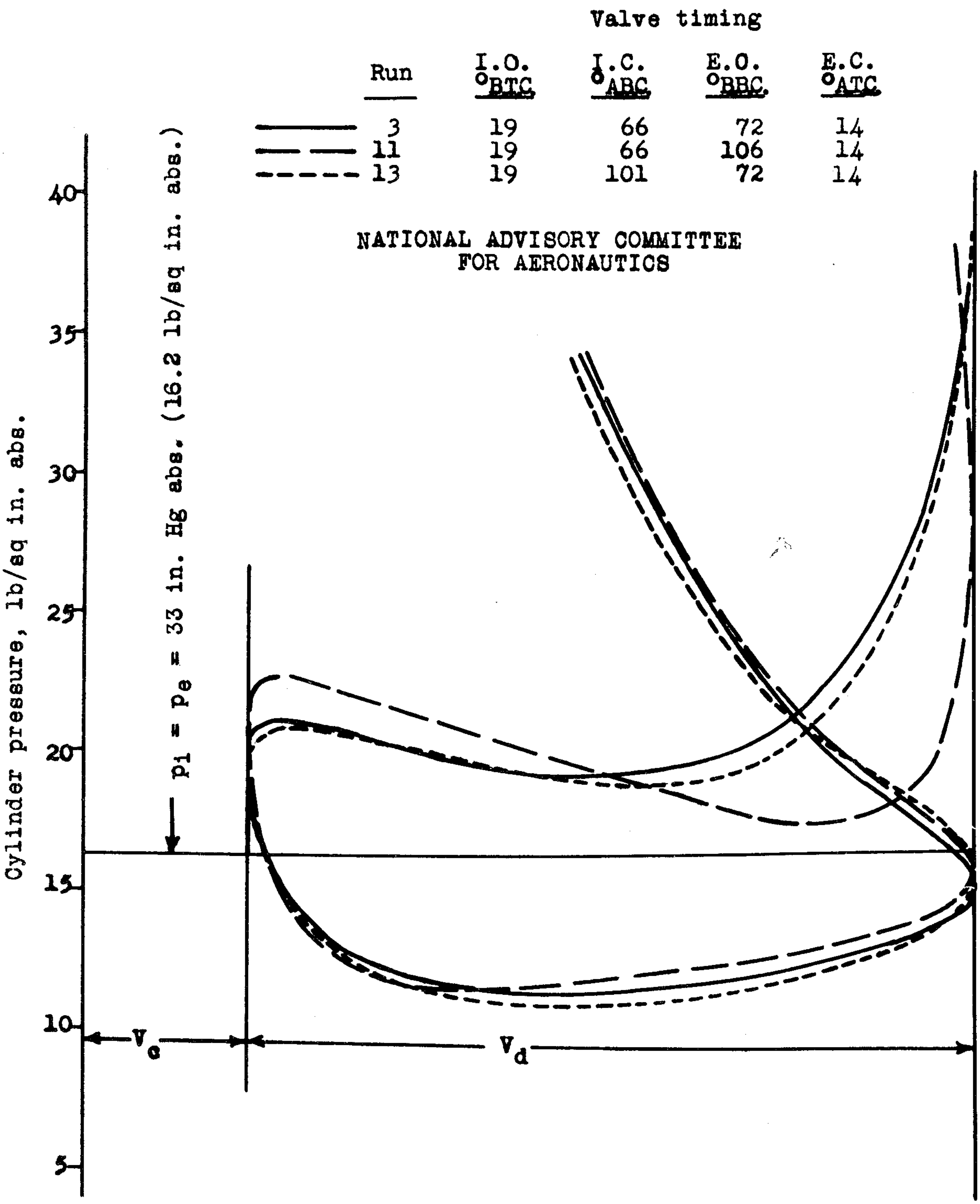

Figure 32.- Light-spring indicator cards. Effect of changing valve timing. Engine speed, 2500 rpm; inlet D^2C_{av}, 1.151; exhaust D^2C_{av}, 1.155; flow-capacity ratio, 1.00.

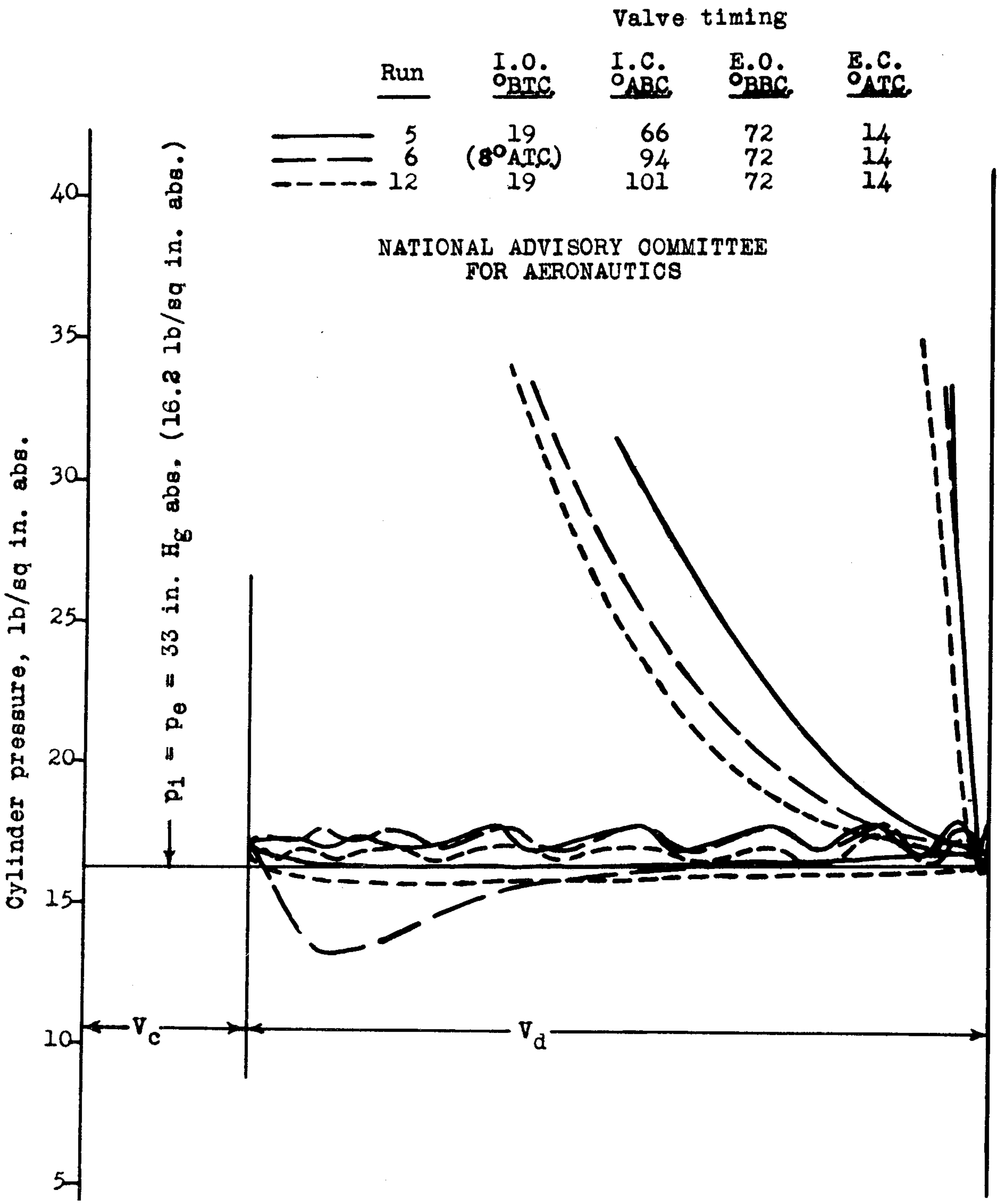

Figure 33.- Light-spring indicator cards. Effect of changing valve timing. Engine speed, 500 rpm; inlet D^2C_{av}, 0.836; exhaust D^2C_{av}, 1.500; flow-capacity ratio, 1.79.

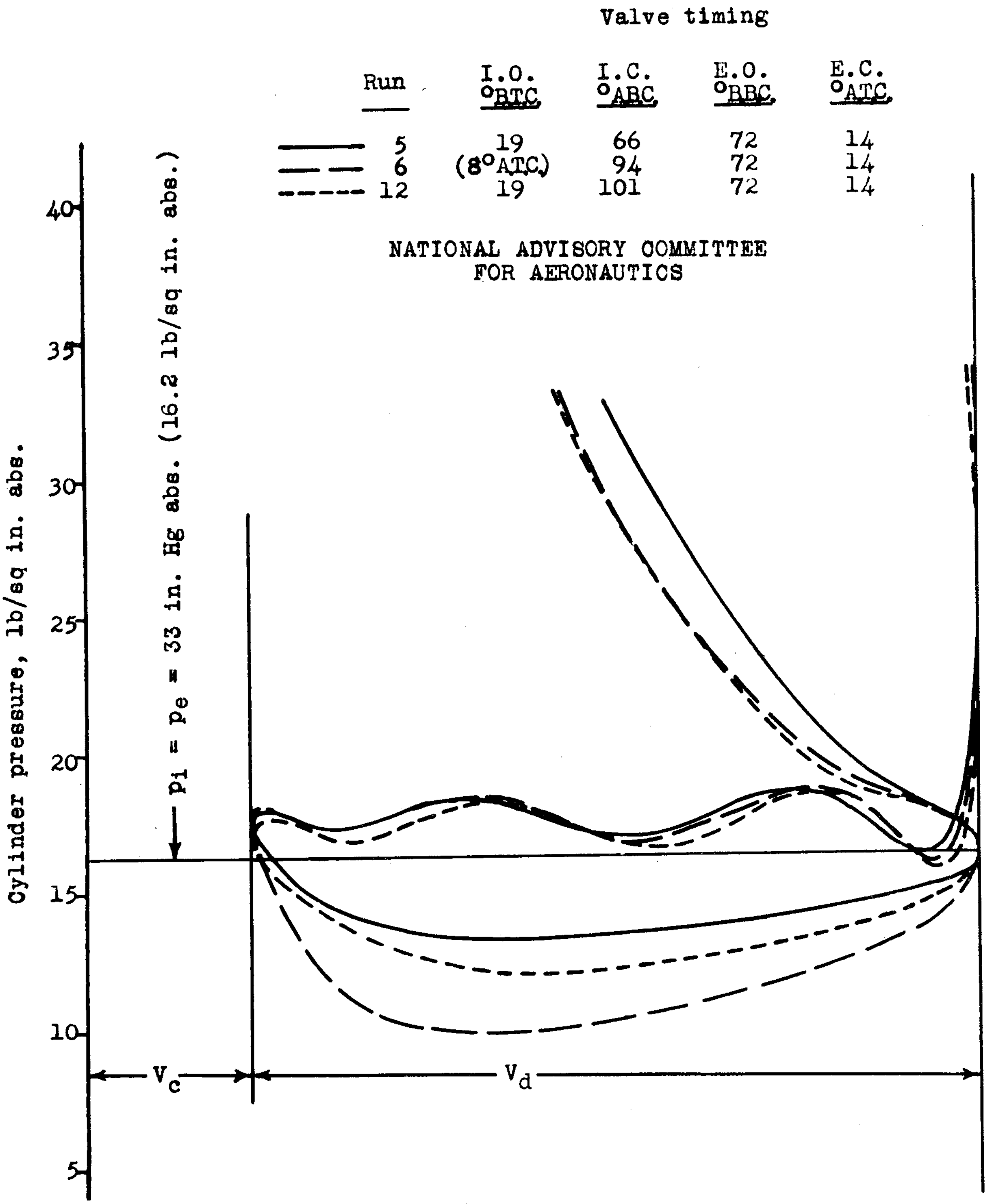

Figure 34.- Light-spring indicator cards. Effect of changing valve timing. Engine speed, 1500 rpm; inlet D^2C_{av}, 0.836; exhaust D^2C_{av}, 1.500; flow-capacity ratio, 1.79.

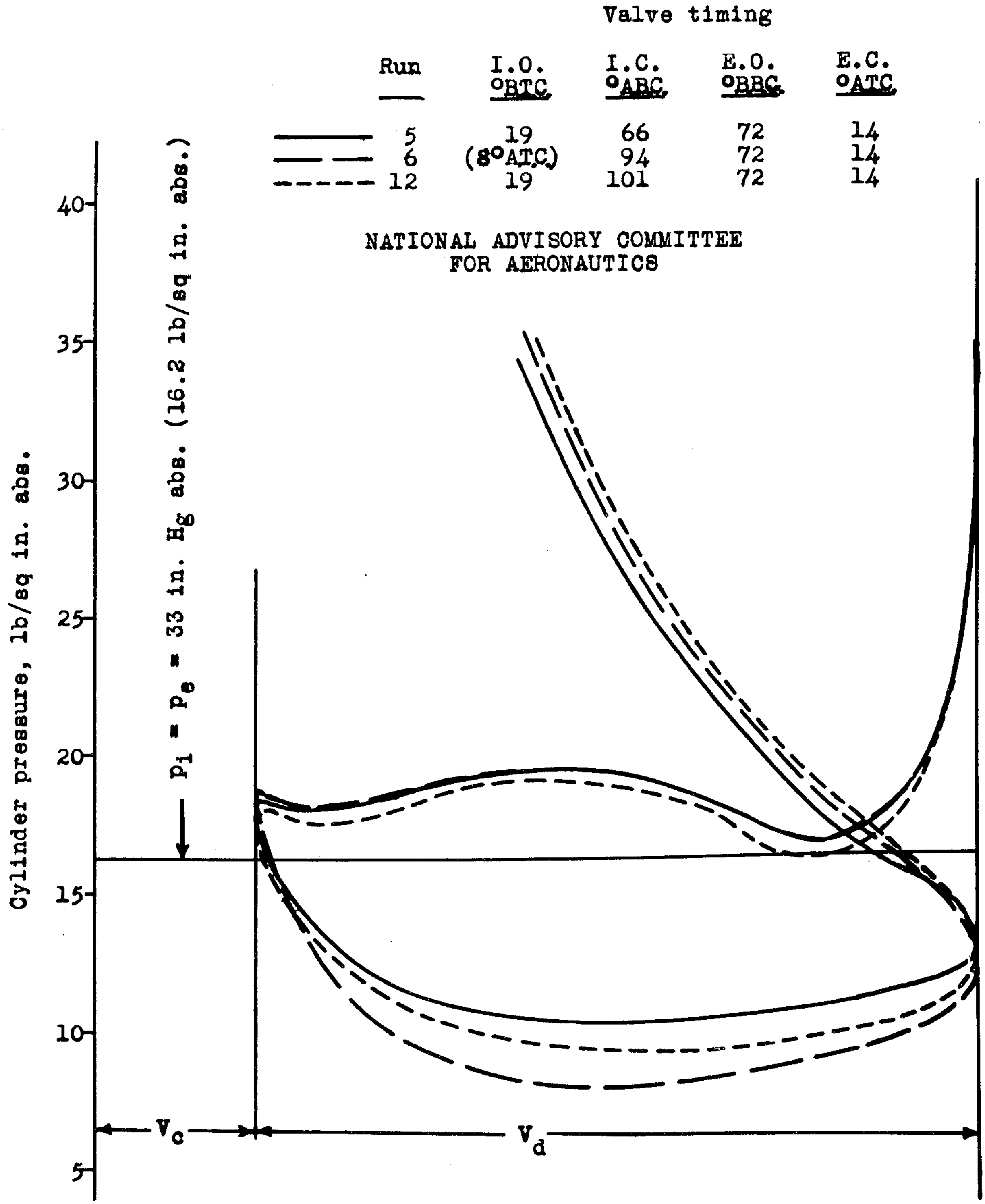

Figure 35.- Light-spring indicator cards. Effect of changing valve timing. Engine speed, 2500 rpm; inlet D^2C_{av}, 0.836; exhaust D^2C_{av}, 1.500; flow-capacity ratio, 1.79.

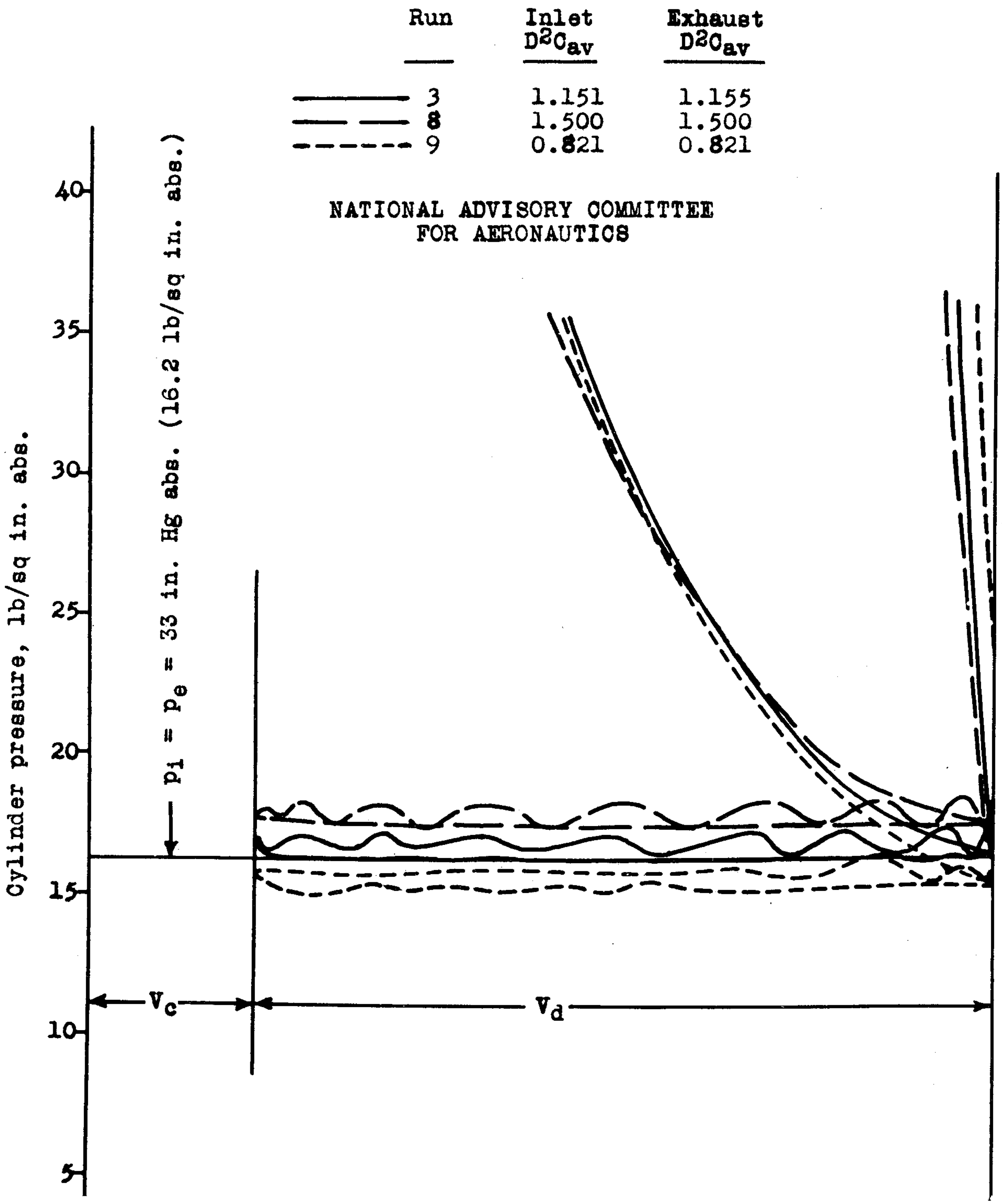

Figure 36.- Light-spring indicator cards. Effect of varying valve flow capacities. Engine speed, 500 rpm; constant flow-capacity ratio, 1.00; normal valve timing (see table I).

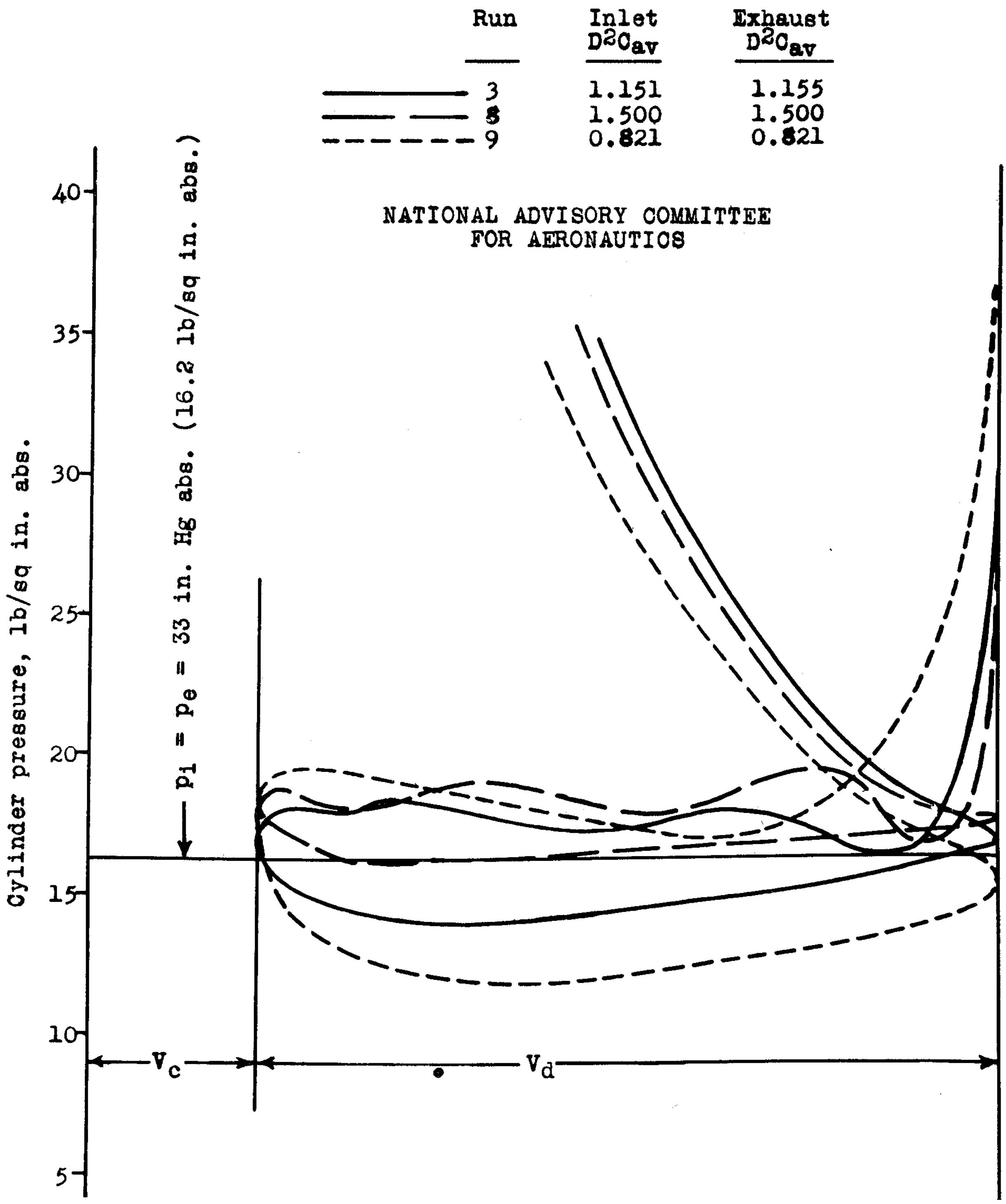

Figure 37.- Light-spring indicator cards. Effect of varying valve flow capacities. Engine speed, 1500 rpm; constant flow-capacity ratio, 1.00; normal valve timing (see table I).

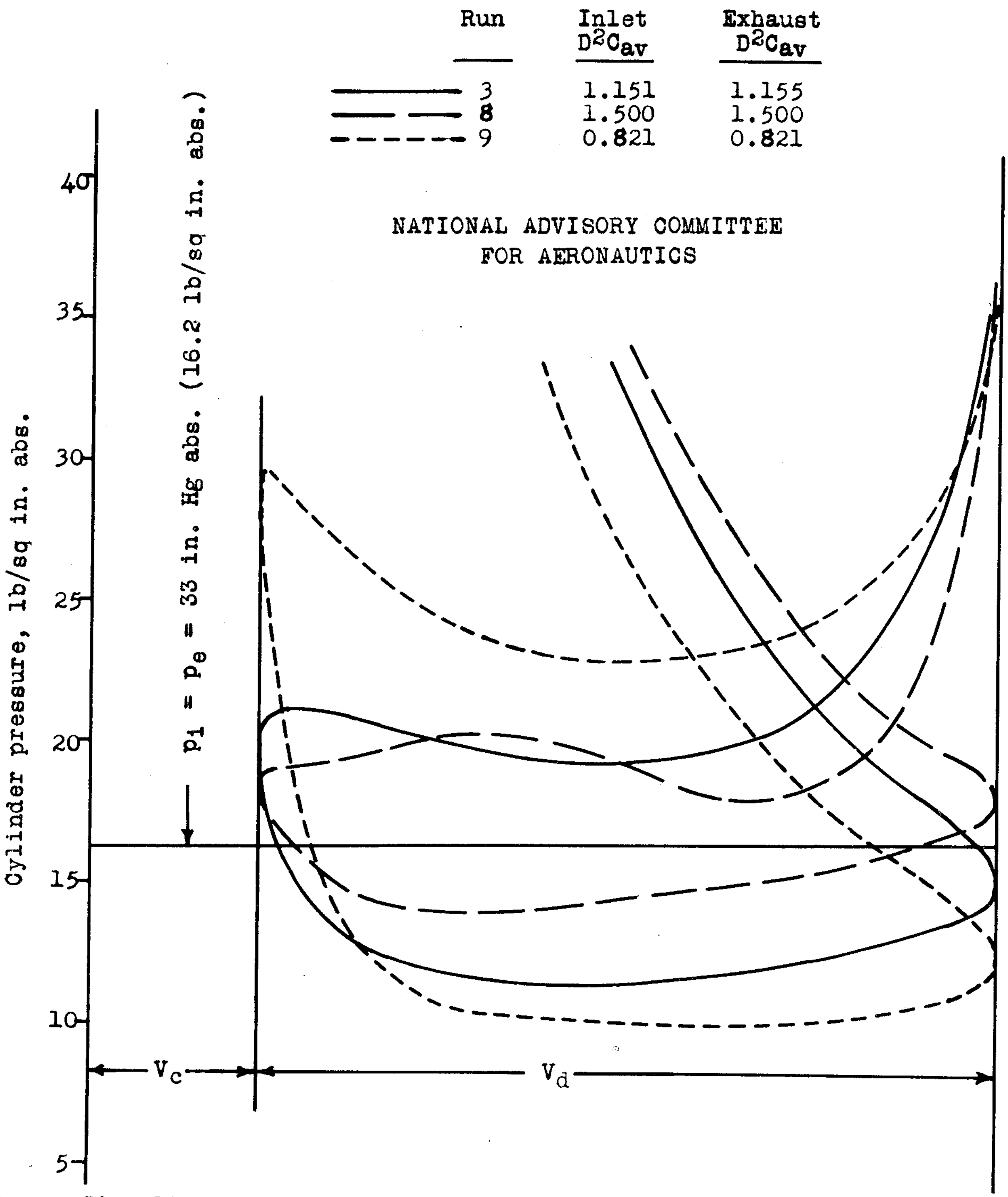

Figure 38.- Light-spring indicator cards. Effect of varying valve flow capacities. Engine speed, 2500 rpm; constant flow-capacity ratio, 1.00; normal valve timing (see table I).

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ABSTRACT

Tests were made with a single-cylinder engine to determine the effect on volumetric efficiency and on engine performance of changing the ratio of exhaust-valve flow capacity to inlet-valve flow capacity when operating with exhaust pressure equal to inlet pressure. It was found that best performance was obtained at ratio of exhaust to inlet-valve flow capacity of approx equal to unity and that highest volumetric efficiency was obtainable with a flow-capacity ratio of 0.69. Tests with specific arrangements of inlet and exhaust valve timing showed no significant improvements in volumetric efficiency.

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